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INFLUENCE OF ARCTIC OPERATIONS ON FUTURE SHIP DESIGN.*

By Capt. Junius W. Millard, U.S.N.R.

The Arctic and sub-Arctic regions are receiving an increasing amount of attention, not only in the public press, but also in our national thinking. This attention stems from a variety of reasons too numerous to be gone into in this paper, but primarily because of the strategic importance of these regions as the locus of great-circle routes between most of the important cities of the world. This strategic importance follows as a corollary to the development of the long-distance airplane.

Man is a gregarious animal, and the history of civilization is the history of settlements expanding east and west due to the pressure

* This article received Honorable Mention in the Society's annual prize essay contest.

of population in the temperate and tropical zones, where food is plentiful and living relatively easy. It was only the restless soul —the non-conformist—who went in a north or south direction from this broad temperate band of civilization, and so the extreme ends of the earth were never thickly settled. This is particularly true in the Antarctic regions. As a result, the number of men who know from experience what living is like in the northernmost and southernmost parts of the world is not only relatively, but also actually, small.

Because of this, as well as lack of interest on the part of the general public, there exists a large amount of misinformation regarding actual Arctic conditions. In the popular mind, this region is one of sub-zero temperatures, constant blizzards, an occasional polar bear and seal, Aurora Borealis, and darkness! It is not the purpose of this paper to point the way out of popular misconceptions, but insofar as open-water Arctic operations are concerned, it must be said that this belief is not based on fact.

From the known facts, however, it is obvious that operations of both land and naval forces in these Arctic regions will require the solution of many problems of logistics and shipbuilding. Particularly is this true in the adaption of present-day ships to such operations. While many structural changes, of course, will be necessary, the greatest emphasis will fall on changes primarily dealing with the comfort and welfare of personnel. In the preparation of this paper an attempt was made to segregate the problems of materiel and personnel, but this had to be abandoned, largely because of their mutual interdependence.

Sea water of average salt-content freezes at approximately 29° F., and so in the open sea this is the lowest water temperature to be encountered. Up to a distance of about forty miles from shore (from the shore ice, that is) the lowest air temperature is about 0° F., and, under this condition of 0° F., winds of about 20-25 knots may be expected. Further, and due to the high vapor pressure of the relatively warm water, the wet and dry bulb temperatures are substantially equal. This low-temperature condition obtains when the wind is blowing to sea from a land mass, but when the direction of the wind is such that it has had a long fetch over open sea (at least 18 hours) the air temperature approaches that of the water, and may be above freezing, even

though forty or fifty miles away temperatures on land may be well below zero. The wit who said that "nothing is permanent but change" might have been speaking of the Arctic regions, for this is one of the most changeable regions of the world as far as weather is concerned.

In addition to the wind and cold, there is probably no other region of the world (with the possible exception of the Antarctic regions) with the high incidence of foggy or cloudy (completely overcast) days, and ruggedly-rough seas—all in all, a navigator's nightmare! With land-marks far inland, and often out of visual range (due to shore ice), and with no celestial observations possible for days on end (due to the overcast), mechanical navigation aids, such as Loran and Radar, must be more highly developed, and almost lavishly provided.

Over the cold inland areas it is generally fairly calm, but as there is nearly always wind over the Arctic water, varying from a low of perhaps 5-10 knots to a maximum of about 100 knots, perhaps it will be well at this time to explain the effect of wind, or air-motion, on well-being. Two methods of empirical comparison are used: "effective temperature" for above freezing temperatures, and what is known as the "wind-chill method" for below freezing temperatures.

Effective temperature, which is used most widely in the band of, roughly, 50° to 90° F., varies with dry bulb, wet bulb, and air motion, and is determined experimentally by means of human subjects. For example, air with a given quantity of moisture, and at a constant velocity, is taken as a base; then the variables of dry bulb, wet bulb, and air motion, are changed one at a time until a condition equal to the base condition of comfort is reached. They are then at the same "effective temperature". It has been found that as air motion increases, a compensatory increase in temperature or humidity, or both, is necessary in order to reach the same "effective temperature". And for constant air conditions of temperature and humidity, an increase in air motion means a lower effective temperature, or feeling of bodily comfort.

At temperatures below freezing the absolute humidity, the actual amount of water vapor in the air, is negligible, so the "wind-chill" method achieves the same end result of comparative bodily comfort in the sub-freezing zone by indicating as a factor the

degree of body cooling as it varies with temperature and air motion. (Detailed information on this very interesting subject is to be found in "Measurements of Dry Atmospheric Cooling in Subfreezing Temperatures" by Siple and Passel.) The influence of wind on bodily comfort must not be overlooked, nor underestimated, in ship operations in Arctic waters. As an example, at a constant air temperature of 30 F., with light cloud cover, a 5 mph wind gave a sensation of "very cool"; 15 mph, "very cold"; and 25 mph, "bitterly cold".

The Arctic wind and low temperatures exact a tremendous toll on body heat. To provide this heat, more fuel in the form of food is essential, and so more refrigerated space on a ship is needed to carry provisions for larger meals. Warming and nourishing foods, such as soup, should be provided for between-meal feeding of watchstanders, and to carry this food to the various stations so that it may be delivered hot, compartmented and insulated food containers are a necessity. For the American sailor this food should be implemented by a plentiful supply of hot coffee, which can also be issued in these same insulated food containers, or in thermos jugs.

Because of the low temperature and high wind velocities, protection must be provided for exposed watch-standing personnel. This will call for special enclosures for lookouts, and for better heating for large-caliber gun-houses. In the case of 20- and 40-mm guns, either special shelters, or local heating for personnel who must remain at the gun, must be provided during periods of general quarters, with possible shelters for the crew during condition watches. This special protection for gun crews, or all exposed watch-standers who are more-or-less confined to one spot, may well take the form of electrically-heated clothing which can be plugged into a power receptacle at the station. It has been found that from 200 to 300 watts per man will provide adequate heating.

In this paper "open-water", rather than "ice" operations will be assumed, although the latter cannot be entirely disregarded. Because of the extreme difficulty of locating drifting mush ice, or the smaller bits of bergs, there exists the constant danger of collision with these small bergs. It follows then, that the ship structure must be strengthened to prevent rupture of the skin by

these collisions, with consequent leakage and flooding. In remembering that the greater part of ice is under water, the greatest amount of protection is necessary for the propellers and rudder. This can be obtained by fairing in the hull and struts to form a sort of tunnel. Fortunately, the presence of large ice floes and bergs is frequently, but not always, made known by a large drop in air temperature, but these smaller "bergy bits" (which are about as large as a small cottage), not only do not announce their presence, but also, because of their closeness to the surface of the sea, cannot be found by other than alert lookouts. "Growlers" are even more dangerous. These are smaller than "bergy bits", and are not only almost the same color as the sea, but are almost even with its surface, making detection extremely difficult. There exists then a great need for mechanical or electronic methods of ice location, particularly when operating relatively near to the shore-ice.

One of the greatest dangers in Arctic operations is the freezing of spray into ice on the topside of a ship. There have been many instances where this has been of such an amount as actually to capsize a ship. For Arctic use a ship must be designed "dry" first of all, and in addition, means must be provided to remove such ice before it assumes dangerous proportions. This may take the form of vibrating tools such as air hammers, steam, hot salt water, chemicals, or manual chipping with wooden mallets or clubs. Of these, hot salt water is probably the most effective and easiest to provide, as it can be distributed about the weather decks by means of the firemain. Heating of this sea water can be either by a central heater in the machinery spaces, or locally by supplying steam to individual fire plugs on the weather decks, although this latter method is wasteful of fresh water unless local heat exchangers are used. Chemicals, because of their corrosive nature, do not seem to have much use in naval vessels to prevent ice formation.

All exposed rotating, or operating, mechanisms should be enclosed to prevent their being rendered inoperative by ice. If this is not possible, special grease, or in extreme cases, special heating devices should be provided. The hawsepipes are a particularly vulnerable part of the ship, especially if there is any probability of having to anchor, and it may be necessary to pro-

vide high-pressure steam coils around them. This will not of itself rid the anchors of ice, but should provide a passage for hot salt water to take care of ice on the anchor itself.

In addition to icing-up of the topside, icing-up below the water-line, oddly enough, must be provided for. In the hull of the ship there are, of course, many sea injections, and these must be placed low enough below the water-line that they are not exposed long enough, or often enough, when the ship rolls that they freeze solid with ice. This is a real danger, particularly in the smaller ships.

While on the subject of ice formation on the topside, in future design of ships for Arctic operation it would appear that greater attention should be paid to better footing in general under icing conditions, and of ways of preventing a man's falling overboard in case of losing his footing. As one method of accomplishing the same result, more interior ways should be provided for passage from one station to another, particularly under general quarters, or blackout, conditions.

Even under the relatively-mild condition of open-sea operation, men cannot survive more than a very few minutes in Arctic water without special warm and waterproof clothing. This special clothing is very bulky, and stowage must be provided for it at the abandon-ship stations in sufficient quantities for the whole crew. Even with these special clothes, men become chilled when in the water, and it may well be that some means of abandoning ship should be provided other than life rafts. This is particularly true for ships operating independently. Any form of abandon-ship boat or raft with more personnel protection than presently provided in the balsa life-rafts calls for stowage provision on an already-crowded ship, as well as an increase in topside weight, as does the stowage of the survival suit. Even with the suit, and in a boat, men cannot long survive in this region, and as a natural result men will not willingly leave the ship as long as she can be kept afloat. In addition, therefore, to strengthening the skin and structure to prevent rupture by ice, greater compartmentation and even better damage control is indicated for ships designed for Arctic operations.

From the foregoing, it will be noted that most of the changes necessary in a ship's structure and layout have to do with the

protection of personnel. Most Navy ships, structurally speaking, are strongly-enough designed for cold-weather operation, or can be made so with a fairly small investment of time and money. But the proper protection of personnel presents a problem so diverse, and with so many ramifications, that much research will be necessary before it is solved satisfactorily. As an example, much thought and time and money has been put into the problem of Arctic clothing. As a result of this the so-called "layer system" has been adopted as the best. This works on the principle of many layers of light, or relatively light, clothes, rather than an equivalent thickness in one or two layers. Successive layers from the outside are removed as the ambient temperature increases, or are added as the ambient temperature decreases. This takes best care of the problem of perspiration, which, strangely, is one of the biggest in Arctic survival. All of this, from a Navy viewpoint, adds up to additional stowage space for the individual sailor as he discards these successive layers. More than that, there is the problem of providing drying space for the clothes which are removed, as well as for the clothes which may become wet by exposure. The problem is complicated by the fact that, as each layer must fit over the one below it, clothing must be handled on an individual, rather than a communal, basis. This is particularly true on a combatant ship where clothes must be readily—immediately—available.

Laundry facilities must be enlarged, and especially equipped, to take care of the large amount of woolens used. Special laundry techniques, and expert supervision, is necessary to prevent undue shrinkage and matting of these woolens.

While on the subject of clothes, a mention of ladders and accesses is germane. Arctic clothes are of necessity more bulky than normal, even foul-weather, gear, and with the heavy shoes and gloves or mittens, the individual man finds the accesses much heavier going. It may be necessary to lessen the rise of ladders, widen and deepen the treads, restrict the use of vertical ladders to a minimum, and widen hatches and doors to accelerate passage. Further, all topside gear that is to be operated by personnel should be designed so that it can be operated by mittened hands; more, by cold, stiff, mittended hands. Light switches, for example, should be of the toggle, rather than the rotating, type; round door knobs,

wherever they cannot be replaced by a lever type of quick-opening door, should be replaced by square door knobs, etc.

Heating and ventilating of living compartments is of prime importance. From an acclimitization standpoint, so that the change from living to working conditions be not too great, and because of the fact that men are wearing heavy underwear, temperatures are carried somewhat lower in these compartments than is usual for winter heating. Clothes may be either damp or wet, and so most of the air should be outside air, preheated to a temperature well above freezing, and then heated to the desired temperature by thermostatically controlled heaters. Adequate exhaust ventilation should be provided to prevent odors, but excess air motion must be carefully guarded against as, at the low room temperature, these would classify as drafts. This difference in the large amount of air for summer use and the small amount for winter use may be handled by either variable-speed, or two-speed, blower motors. Because of the fact that Arctic (sub-freezing) air is, to all practical intents and purposes, bone-dry, and heating still further lowers the relative humidity, the air must be moistened by steam, or water, spray to raise the humidity. Otherwise, drying and irritation of the mucous membranes of the nose and throat will result, with a high incidence of respiratory diseases. If a steam spray is used, care must be taken that unpleasant odors, such as from boiler compounds, are removed before use, and the humidity should be controlled by humidistats. Room odors, because of the relatively small volumes of air employed, are a problem, and may be taken care of by ozonators, or other electric, or chemical, means. To conserve on heat, deliberately over-heated air-locks with quick-opening doors, and these doors opening aft and out of the wind-stream, should be provided at major accesses to the weather decks.

When the navigating bridge is buttoned up and heated for cold-weather conditions, moisture from the breath of the bridge personnel causes the humidity to rise, and this may go up to such an extent that fog or condensation occurs on the bridge windows. This can be overcome by double glass on the windows, by blowing heated air across the window, by dehumidification, or by any other method, which will prevent the window temperature and bridge dewpoint from coinciding, which is necessary for condensation.

For practically the same reason, additional insulation may be necessary on the firemain, bulkheads below the waterline, and similar locations throughout the ship, to prevent sweating and consequent dripping.

The Arctic seems to have been designed deliberately by Providence to discourage its settltement; even, in fact, to encourage its isolation. And yet the Esquimaux, by following certain inexorable laws of food, heating, and clothing, manage not only to exist, but also to lead fairly happy lives. Operation of Naval ships in the Arctic under winter conditions will never be the idyllic existence of life in tropical, or sub-tropical, waters. But the very recognition and admission of the dangers inherent in Arctic operations serves to lessen their terrors to the uninitiated—after that, intelligent perseverance, keeping in mind always the fundamentals of existence, will in time bring Arctic operations to the level of winter patrols in the North Atlantic.

AN APPARENTLY NEGLECTED PHASE OF NAVAL RESEARCH.

By Captain James E. Hamilton, U.S.N., Member.

The destroyer of World War I cost about a million and a quarter dollars and required a crew of about 8 officers and 125 enlisted men. The prototype of World War II cost about eight times as much and requires twice as many officers and three times as many enlisted men. The operating cost is several times as much as the earlier class.

The first cost when converted to equivalent dollars reduces to about four to six times as much for the later class. Thus it is safe to say that both the first and operating costs of a destroyer have more than tripled in about a generation. Nothing in war experience nor post war plans indicates that the Navy can live effectively with a smaller number of destroyers now than it could a few years ago. If the number is assumed to be about the same, the cost of the destroyer arm of the Navy has increased more than threefold, after taking into account the variation in the value of the dollar.

The increased cost is due to a combination of greater size and advanced technology. Both are incident to a combination of research and development on our part and a counter to research and development on the other fellow's part.

The general statement about destroyers is applicable in almost the same degree to every other material element which goes to make up the Navy. The apparent reasons and the trend are well known.

There is a constant demand for improved basic performance, for addition of improved weapons and advanced defensive features. In a technical sense there is no simple point by point comparison between a modern unit and one of a few years before. The natural advance is not rapid enough so there is increased pressure to raise the tempo of research, then development, operational evaluation and finally adoption.

There can be absolutely no objection to the ends sought—that

every naval element, ship or otherwise, should be positively capable of outdoing the best counterpart any competing Navy could put against it. No second best ship can be a part of a best Navy. Yet quality in the unit cannot fully take the place of quantity. No matter what the type or class of a plane or ship, there must be enough of them for the strategical coverage required.

Now of course straight arithmetic brings us to the crux of the matter. How long can any first class country continue to maintain an adequate Navy if the number of units does not decrease when both the first and operating costs of each unit constantly increase? Resistance against current costs is already great. There can be a breaking point when the Treasury cannot take it or the Congress will not let it try. The task which research and development should be given is to start a downward price and cost trend before the breaking point is reached.

The fruit of research is something to be applied to a practical function. The translation requires spending money for material and labor for converting that material into the ultimate form required. On the side line from mine or forest to finished product are stockholders awaiting dividends, inventors awaiting royalties, entreprenuers awaiting profits and the like. There is nothing wrong with any of these elements to which the Treasury's money is distributed. The objection is, if there is any, that there are too many of them, and the reason seems to be the path followed by Naval material development.

Limiting the discussion to ships, the first demand is for increased size in the new prototypes. Increased size is demanded for better sea keeping qualities and to provide carriage for more things. Increased size alone does not increase the price markedly. At any one time, the cost per ton of naval vessels decreases as the tons per ship increases. Doubling the size of a class of ship might increase the cost of the unit by 25 per cent. No fault can be found with that. If a bigger ship is needed, and that decision is easily reached as a strictly technical-naval matter, anyone would expect to pay more for it. However, when the cost per ton of the ship over a period of about 20 years triples in equivalent dollars, the increase can be of some concern.

One cites increased costs of labor and material as basic reasons. Of course that is the major reason but the answer isn't just that

a ton of steel costs more or that an electrician gets more for each hour of work. Those increases disappear in the evaluation of the dollar. The reason is the requirement for more expensive kinds of material and for the use of higher grades of labor and more of it to work each ton of raw material into a finished ship. The reason for both is the course which technological advance has taken. The trend is marked with gadgetry.

Simply a naval vessel is a steel hull provided with propelling machinery and then equipped with hotel services for her crew and a multitude of weapons, communication and detection devices and filled with food, fuel, ammunition, operating supplies and spare parts. The hull, the propelling machinery, and some of the weapons and other equipment have had functional counterparts for many years. Other parts of the equipment are being added continuously as new things are developed. It could be expected that a new functional component added to a ship would increase the cost by the cost of that component. Also if the number of installed units of any functional component is increased the cost would naturally increase. However, it could be expected that as functional components are improved, the unit cost of some of them might decrease or at least a better gadget could be bought for the same money. Such seems rarely to be the case.

Any general statement on this subject can be challenged for cause. Nevertheless, I shall venture the following:

(a) Instead of working to adopt cheap, common basic materials to accomplish a purpose or to devise a means of using that material for the purpose, the tendency is to go to materials which are rarer, or contain rare and costly ingredients or require more, or more costly labor to fabricate.

(b) There is a tendency to adopt a thing (continuous use of the term "gadget" may be upsetting to some readers) which has been separately invented and patented to the end that many advanced devices are largely complicated assemblies of many things, each of which demands separate royalties and other management and entreprenurial overhead.

(c) The preceding tendency leads to devices which require a relatively large operating crew.

(d) Automatic devices are developed and adopted but their reliability is such that one, or more, stand-bys is always retained.

(e) Possibility of failure is so real that specially trained maintenance crews must be available.

(f) Construction is so intricate and special that repairs cannot be effected by a reasonably equipped shop and a stock of basic materials. Many costly, tailor-made spare parts must be carried on board and also in the logistic supply system.

The results of such tendencies in research and development are, among others:

(a) Existing ships must be radically changed for adoption of a new device and they become obsolete much faster because they reach the point where additional weight and added crew accomodations cannot be accepted.

(b) New ships get bigger and bigger and require larger and larger crews.

(c) Because of a combination of larger crews and rapid advance of standards of living, hotel accomodations become more difficult to install and may themselves force a secondary increase in the size of the ship.

(d) The supply system becomes more and more complicated. A myriad of new items must be carried and technical store keeping gets more and more expensive.

(e) Training and education of all personnel becomes much more expensive and extensive. More time of personnel is spent in training with the result that the percentage, of those available, which are available for operating billets is reduced markedly. There is even a possibility that, if the trend continues, it will be difficult to find sufficient people with capacity to absorb the necessary training.

It may be cited in rebuttal of this article that research and development in main propelling machinery has increased the efficiency so that greater speeds for longer hours with less fuel are now attained. True and truly praiseworthy, yet the total spent for fuel these days is only about one thirtieth as much as is being spent for pay and allowances of officers and enlisted men.

This cannot be a constructive article in a specific way. The improvement which must be sought for can be found only by

ingenious research and development. First the need must be recognized and not only recognized but seen in its really true proportions.

The Navy is essential to the continuation of the United States. I believe that to be true for as long as any living being continues to enjoy life and even though the United Nations is 100 per cent successful. Maintenance of power short of war is a matter of negotiation which means compromise. Most compromises can be accepted but occasions will arise when neither party can conscientiously give in. When these situations develop war will be avoided only if one party is so strong and ready that common sense tells the other side to back down. I am not alone in the hope that the United States will always have a clear international conscience and to protect it will not have to resort to war or to surrender principles to expediency.

Most of the people in the United States will probably agree to maintaining an adequate Navy. They really have to depend on the Navy itself to tell them what is adequate. They would probably accept that if it wasn't accompained by such a staggering bill when measured in dollars. Even a householder, in his relatively simple affairs, will give up something basic when the hand of the sheriff gets too close to his shoulder.

We are in a state now where people are still disturbed by the recent war and are a little uncertain about demanding too much of a squeeze on naval appropriations. However, we can expect that from year to year the uncertainty will decrease and the size of taxes and of the national debt will become increasingly more important in their thinking than the need for naval appropriations. It will be ineffective to rant about "an ounce of prevention", etc. Certainly a war will be expensive. It might break the national economy. But we have nothing but history to point to the probability of another war. We want to live today and not under the shadow of a straining Federal Treasury. History proves more than the probability of another war. It points unerringly to the fact that each year the Navy can expect fewer operating dollars until a proveable bottom is reached or until the war's threat is apparent to all.

The Navy can expect some relief from a more valuable dollar. What must be done is to make the relative gain of the Naval

dollar much greater than that of the commercial dollar. We must develop our concepts and our materials and our gadgets so that ships and airplanes are cheaper to buy, although first cost is definitely of secondary importance.

Of far greater importance, we must develop and design and procure airplanes and ships so that the numbers we require for implementation of reasonable strategic plans can be manned adequately with considerably fewer people who require less complicated training and they must be and remain operable with a far simpler and cheaper supply system.

The goal of research and development must necessarily continue to be to give this country the very best munitions. By "best" they must be capable of inflicting more damage on an enemy objective and of withstanding more punishment than the other fellows can. With that end attained as a first step, progress must not be stopped until the item has been improved to a maximum of simplicity, ruggedness, and reliability and to the point where it can be operated with full effectiveness with a minimum crew of moderately trained Americans.

Research and Development have carried the Navy far and at that have just reached the frontier of advance. They must continue forward but from now on must be able to point with pride to an accomplishment of protecting the Treasury concurrently with furnishing conventional naval security.

GAS TURBINE PERFORMANCE CHARACTERISTICS, DESIGN CONSIDERATIONS, AND APPLICATIONS.

By J. T. Rettaliata*.

Introduction.

Practically all present day gas turbines operate on a cycle wherein combustion occurs essentially at constant pressure instead of at constant volume, as in the case of the Holzwarth turbine. Those operating on the former cycle are referred to as combustion gas turbines. Of these, the majority are of the "open" cycle type where air is continuously taken in from the atmosphere and exhaust gases from the turbine are discharged back to the atmosphere. There are other gas turbine cycles in existence which are referred to as "closed" cycles because the working medium is not expelled from the system but, rather, is continuously recirculated.

Basic constituents of a combustion turbine unit are a gas turbine, compressor, combustion chamber, and fuel equipment. Supplementing these with a generator, lubrication system, control apparatus, and a means for starting, completes a power plant significant primarily because of its lack of the veritable myriad of auxiliaries necessary for steam plant operation. Furthermore, since its working medium, air, is universally available and because water is not required in its operation, the basic gas turbine plant enjoys the added unique advantage of being independent of geographical location.

The arrangement of a modern combustion gas turbine unit is depicted in Fig. 1. Air from the atmosphere is compressed in a multi-stage axial flow compressor B, driven by a reaction type gas turbine A. Liquid or gaseous fuel injected at C is burned with part of the air discharged from the compressor; the remaining and greater part flows through the annular space F and, upon emerging, mixes with and cools the products of combustion to a suitable turbine inlet temperature. Expansion of this gas to atmos-

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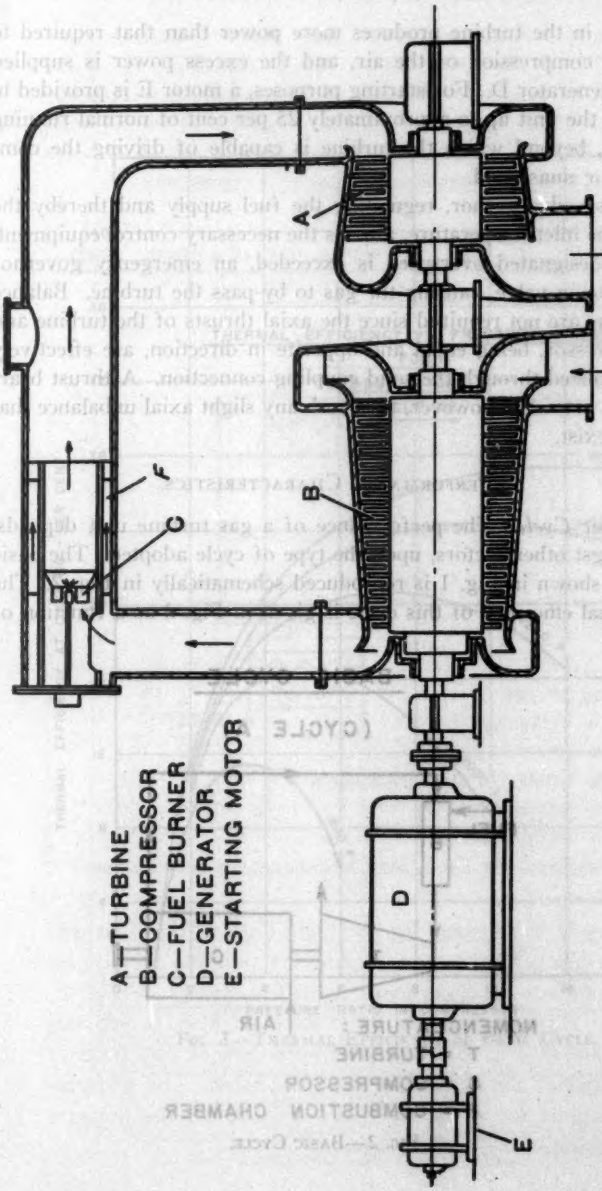


FIG. 1.—SECTION THROUGH COMBUSTION GAS TURBINE-AXIAL COMPRESSOR UNIT.

phere in the turbine produces more power than that required to effect compression of the air, and the excess power is supplied to a generator D. For starting purposes, a motor E is provided to bring the unit up to approximately 25 per cent of normal running speed, beyond which the turbine is capable of driving the compressor unassisted.

A speed governor, regulating the fuel supply and thereby the turbine inlet temperature, affords the necessary control equipment. If a designated overspeed is exceeded, an emergency governor actuates a valve, causing the gas to by-pass the turbine. Balance pistons are not required since the axial thrusts of the turbine and compressor, being equal and opposite in direction, are effectively neutralized through the solid coupling connection. A thrust bearing is provided, however, to absorb any slight axial unbalance that may exist.

Performance Characteristics.

Basic Cycle—The performance of a gas turbine unit depends, amongst other factors, upon the type of cycle adopted. The basic cycle shown in Fig. 1 is reproduced schematically in Fig. 2. The thermal efficiency of this cycle is given in Fig. 3 as a function of

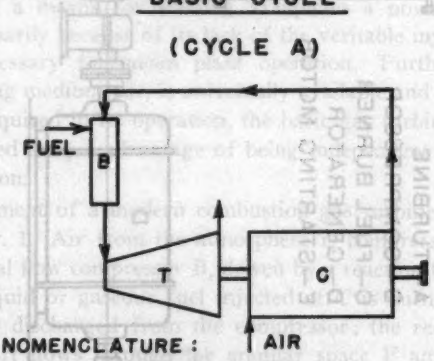


Fig. 2.—Basic Cycle.

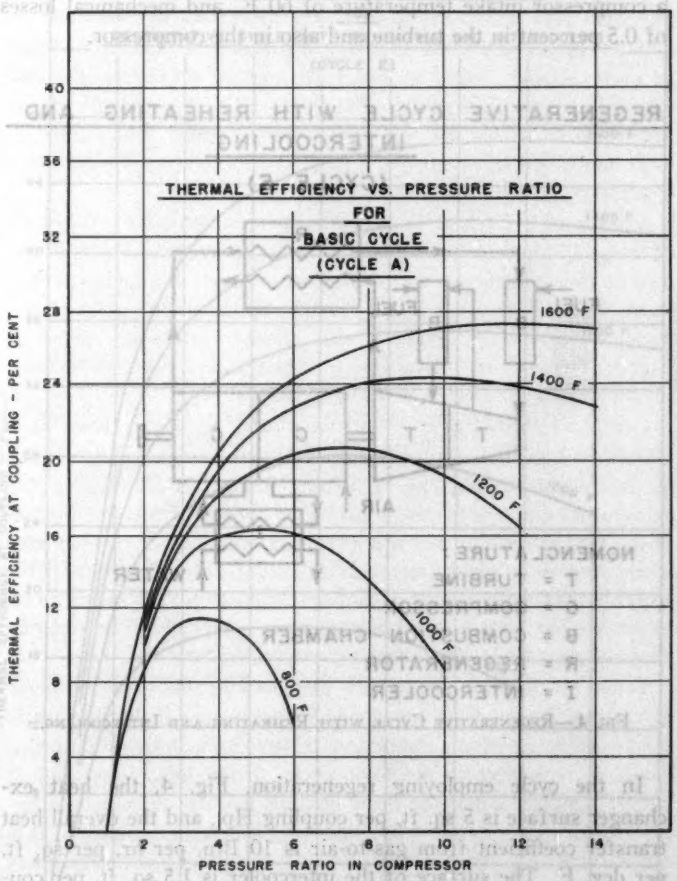


FIG. 3.—THERMAL EFFICIENCY OF BASIC CYCLE.

pressure ratio for various turbine inlet gas temperatures from 800° F to 1600° F. The performance data presented in Figs. 3 and 5 are based on a turbine efficiency of 86 per cent, a compressor efficiency of 84 per cent, a combustion efficiency of 98 per cent, a compressor intake temperature of 60 F., and mechanical losses of 0.5 per cent in the turbine and also in the compressor.

REGENERATIVE CYCLE WITH REHEATING AND INTERCOOLING

(CYCLE E)

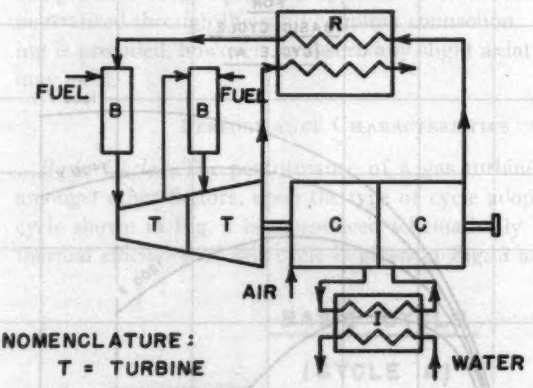


FIG. 4.—REGENERATIVE CYCLE WITH REHEATING AND INTERCOOLING.

In the cycle employing regeneration, Fig. 4, the heat exchanger surface is 5 sq. ft. per coupling Hp., and the overall heat transfer coefficient from gas-to-air is 10 Btu. per hr. per sq. ft. per deg. F. The surface of the intercooler is 1.5 sq. ft. per coupling Hp., and an overall water-to-air coefficient of heat transfer of 20 Btu. per hr. per sq. ft. per deg. F. is taken. The performance calculations for all of the cycles account for the pressure drops occurring in the various elements.

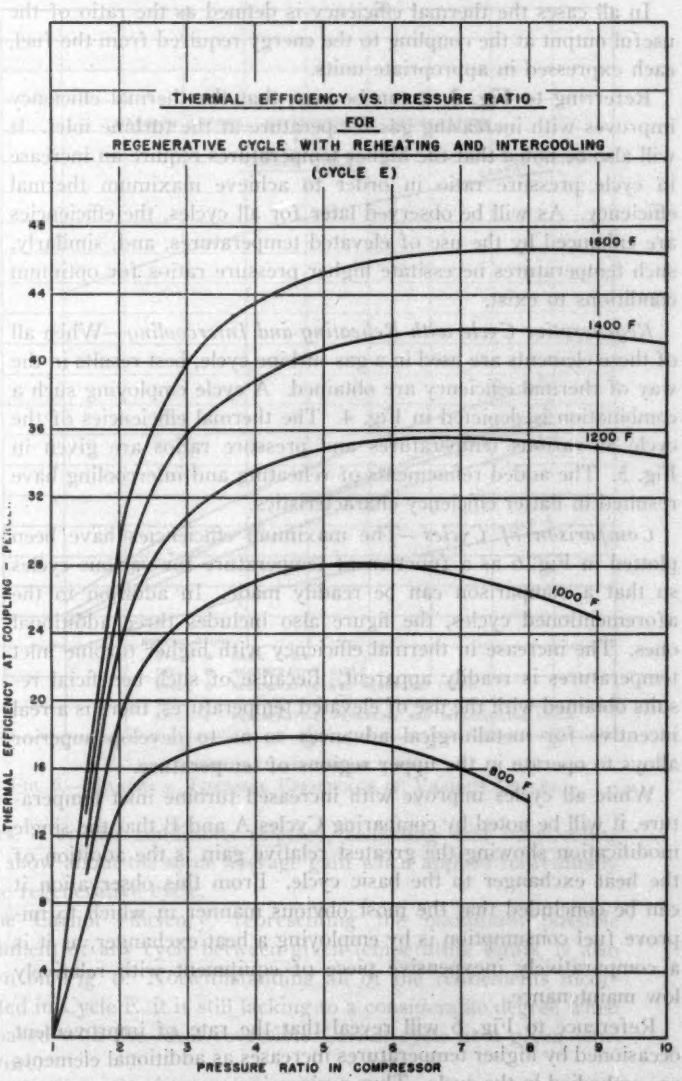


FIG. 5.—THERMAL EFFICIENCY OF REGENERATIVE CYCLE WITH REHEATING AND INTERCOOLING.

In all cases the thermal efficiency is defined as the ratio of the useful output at the coupling to the energy required from the fuel, each expressed in appropriate units.

Referring to Fig. 3, it can be seen that the thermal efficiency improves with increasing gas temperature at the turbine inlet. It will also be noted that the higher temperatures require an increase in cycle pressure ratio in order to achieve maximum thermal efficiency. As will be observed later for all cycles, the efficiencies are enhanced by the use of elevated temperatures, and, similarly, such temperatures necessitate higher pressure ratios for optimum conditions to exist.

Regenerative Cycle with Reheating and Intercooling—When all of these elements are used in a gas turbine cycle, best results in the way of thermal efficiency are obtained. A cycle employing such a combination is depicted in Fig. 4. The thermal efficiencies of the cycle at various temperatures and pressure ratios are given in Fig. 5. The added refinements of reheating and intercooling have resulted in flatter efficiency characteristics.

Comparison of Cycles—The maximum efficiencies have been plotted in Fig. 6 as a function of temperature for various cycles so that a comparison can be readily made. In addition to the aforementioned cycles, the figure also includes three additional ones. The increase in thermal efficiency with higher turbine inlet temperatures is readily apparent. Because of such beneficial results obtained with the use of elevated temperatures, there is a real incentive for metallurgical advances so as to develop superior alloys to operate in the upper regions of temperature.

While all cycles improve with increased turbine inlet temperature, it will be noted by comparing Cycles A and B that the single modification showing the greatest relative gain is the addition of the heat exchanger to the basic cycle. From this observation it can be concluded that the most obvious manner in which to improve fuel consumption is by employing a heat exchanger, as it is a comparatively inexpensive piece of equipment with relatively low maintenance.

Reference to Fig. 6 will reveal that the rate of improvement occasioned by higher temperatures increases as additional elements are embodied in the cycle. Thus a given increment in temperature benefits the more complex cycles to a greater extent than the

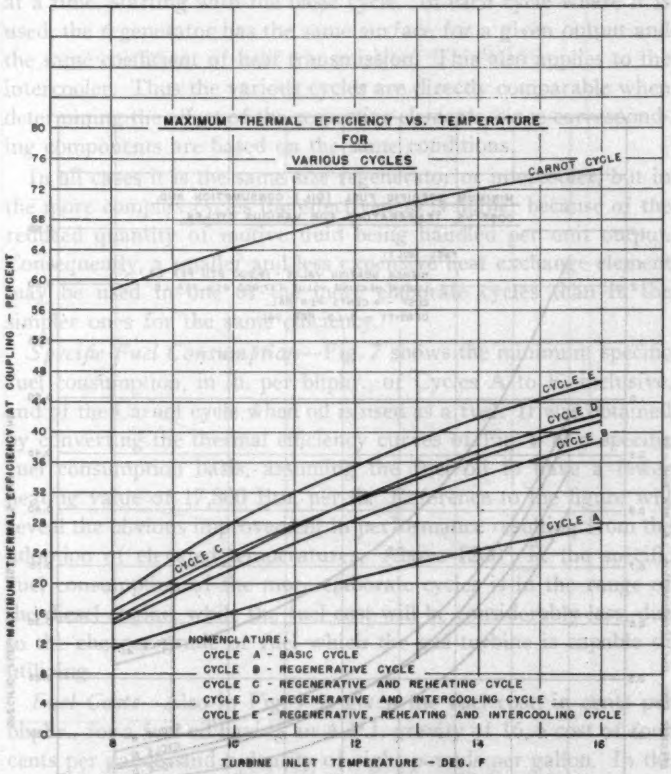


FIG. 6.—MAXIMUM THERMAL EFFICIENCY OF VARIOUS CYCLES.

others. For the conditions selected, reheating and intercooling each show about the same average gain when applied individually to the regenerative cycle.

The Carnot efficiency, representing the maximum possible attainment of any cycle between given temperature limits, is also shown on Fig. 6. Notwithstanding all of the refinements incorporated in Cycle E, it is still lacking to a considerable degree when compared with the ideal reversible Carnot cycle at a given temperature.

In the foregoing cycles, it was intended to show the effect on performance of a change in the cycle by adding a single element

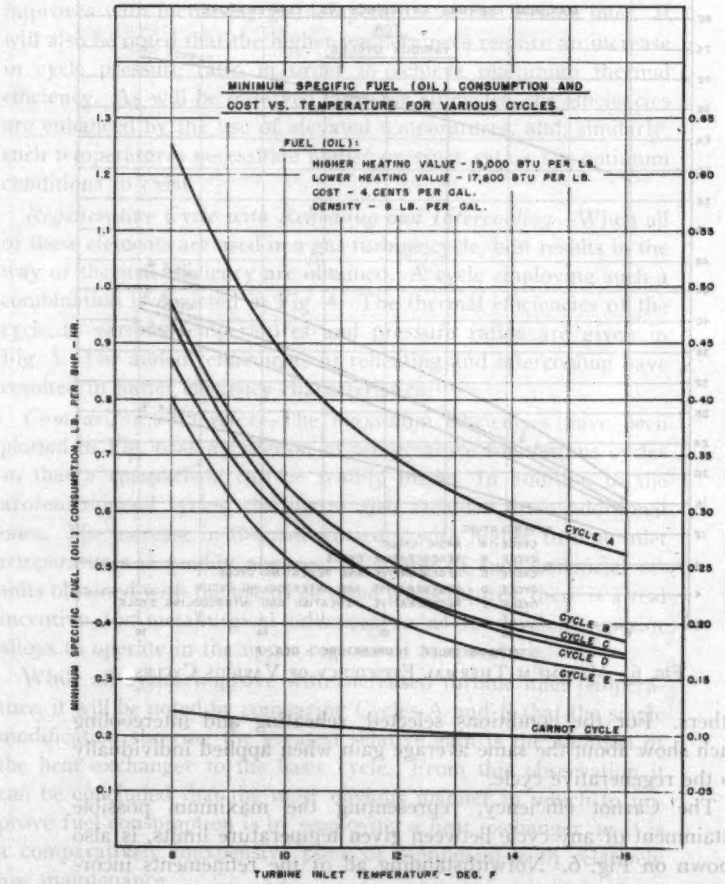


FIG. 7.—MINIMUM SPECIFIC FUEL CONSUMPTION AND COST FOR VARIOUS CYCLES USING OIL FUEL.

at a time, starting with the basic cycle. In each cycle where it is used, the regenerator has the same surface for a given output and the same coefficient of heat transmission. This also applies to the intercooler. Thus the various cycles are directly comparable when determining the effect of the respective elements, since corresponding components are based on the same conditions.

In all cases it is the same size regenerator or intercooler, but in the more complex cycles the effectiveness is higher because of the reduced quantity of motive fluid being handled per unit output. Consequently, a smaller and less expensive heat exchange element may be used in one of the more elaborate cycles than in the simpler ones for the same efficiency.

Specific Fuel Consumption—Fig. 7 shows the minimum specific fuel consumption, in lb. per bhphr., of Cycles A to E, inclusive, and of the Carnot cycle when oil is used as a fuel. It was obtained by converting the thermal efficiency curves of Fig. 6 to a specific fuel consumption basis, assuming the fuel oil to have a lower heating value of 17,800 Btu. per lb. Reference to the figure will reveal the obvious improvement in performance resulting from the adoption of elevated temperatures. Above 1200° F. the specific fuel consumption of the more elaborate cycles is in the range of the diesel engine, while the fuel cost will be considerably less, due to the cheaper grade of fuel which the gas turbine is capable of utilizing.

Fuel Costs—Also in Fig. 7 is given the fuel cost, in cents per bhphr., for a fuel oil having an A.P.I. gravity of 16, a cost of four cents per gallon, and a density of eight pounds per gallon. In the higher temperature regions the cycles employing regeneration and other means for increasing efficiency are attractive on a fuel cost basis, especially when compared with other prime movers using liquid fuels.

Optimum Pressure Ratio—Although the pressure ratio at which maximum efficiency occurs in a given cycle is not precisely a linear function of temperature, it may be approximated as such, as shown in Fig. 8. It can be seen that the greatest variation in optimum pressure ratio with temperature occurs in the basic cycle. Thus, to build such a cycle for maximum performance at elevated temperatures would necessitate many stages in the turbine and compressor. By adopting regeneration (Cycle B), these elements

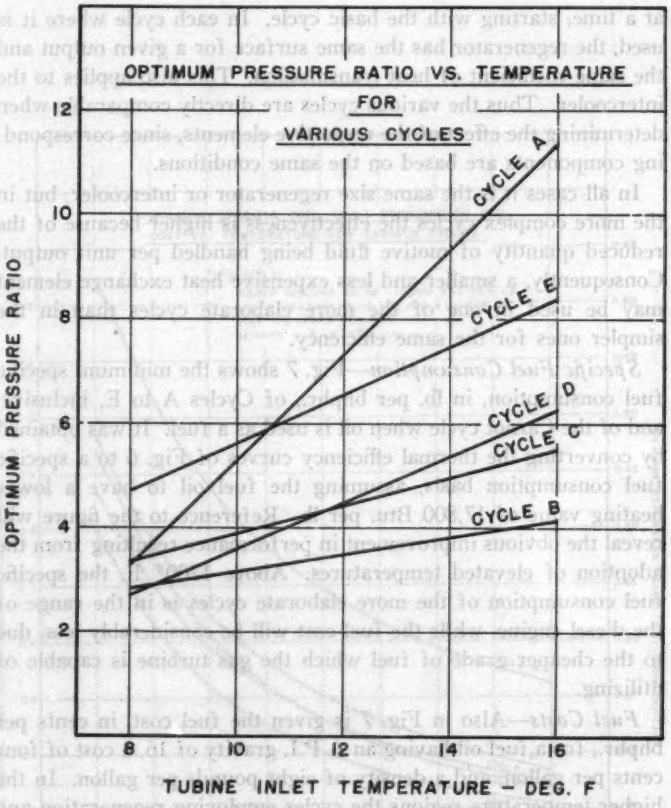


FIG. 8.—OPTIMUM PRESSURE RATIOS FOR VARIOUS CYCLES.

can be simplified, due to the lower pressure ratio required, and higher efficiency can be achieved at a given temperature. Except in some special applications which may justify it, it is unlikely that the basic cycle would be constructed for peak performance at high temperatures.

Reference to Fig. 8 reveals that the adoption of reheating or intercooling, individually or in combination, necessitates higher pressure ratios for the attainment of optimum conditions. Above a turbine inlet temperature of 1050° F., however, even with Cycle E, these pressure ratios are still less than the basic cycle requires.

Performance at partial load—In general, maximum thermal efficiency at partial load can be achieved by maintaining the turbine inlet temperature at a high value. Optimum conditions at reduced load ensue when the temperature is kept constant at the design value for all loads. In order to accomplish this in open cycles at reduced output, it is necessary to diminish the flow of the working substance, which requires variable compressor speed. Where the nature of the load is such that constant rotative speed is not necessary, then the variable flow requirement can be fulfilled even in cycles comprising only a single shaft with one turbine and one compressor, thereby making possible the attainment

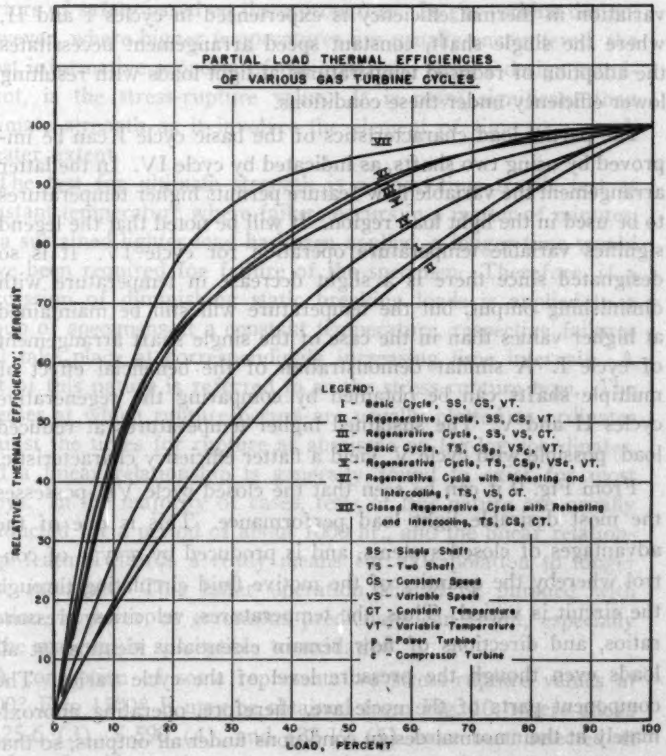


FIG. 9.—THERMAL EFFICIENCY AT PARTIAL LOAD FOR VARIOUS CYCLES.

of good partial load efficiencies. If the load is of a fixed speed type, such as an alternating current generator operating at constant frequency, in order to achieve variable flow a multiple turbine arrangement is necessary where one turbine drives the generator, and the other, the compressor.

The relative thermal efficiencies of various gas turbine cycles are given in Fig. 9 as a function of load. It can be seen that the flattest characteristics are obtained in cycles III, VI, (1) and VII (2), where the turbine inlet temperature is maintained essentially constant at the normal design value at all loads. In order to accomplish this, it will be noted from the legend that variable speed operation is employed in the open cycles. The greatest variation in thermal efficiency is experienced in cycles I and II, where the single shaft, constant speed arrangement necessitates the adoption of reduced temperatures at light loads with resulting lower efficiency under these conditions.

The partial load characteristics of the basic cycle I can be improved by using two shafts, as indicated by cycle IV. In the latter arrangement the variable flow feature permits higher temperatures to be used in the light load region. It will be noted that the legend signifies variable temperature operation for cycle IV. It is so designated since there is a slight decrease in temperature with diminishing output, but the temperature will still be maintained at higher values than in the case of the single shaft arrangement of cycle I. A similar demonstration of the beneficial effect of multiple shafts can be obtained by comparing the regenerative cycles II and V. The sustained higher temperatures at reduced load, possible with cycle V, yield a flatter efficiency characteristic.

From Fig. 9 it can be seen that the closed cycle VII possesses the most desirable light load performance. This is one of the advantages of closed systems, and is produced by a type of control whereby the quantity of the motive fluid circulating through the circuit is varied. Thus, the temperatures, velocities, pressure ratios, and directions of flow remain essentially identical at all loads even though the pressure level of the cycle varies. The component parts of the cycle are, therefore, operating approximately at their normal design conditions under all outputs, so that the efficiency is maintained at a high value even at light loads.

Design Considerations.

As has been previously observed, the performance of gas turbines improves with the use of elevated temperatures. Consequently, the necessity for adopting suitable materials to withstand high temperature service is manifest. Progress in the gas turbine field will always be contingent upon metallurgical advances, and real incentives exist for the development of alloys capable of operating at extreme temperature limits.

In steam turbine practice, various properties of the materials employed have been used as a basis for design. In particular, the ultimate, yield, and creep strength values have been found useful in determining the proportions of structures having suitable factors of safety based on these properties. In gas turbine design, however, where higher temperatures are usually encountered, the most informative property of the materials, from a design standpoint, is the stress-rupture value. It is more significant than ultimate strength as it involves the element of time to a much greater extent.

The test for ultimate strength is a short-time tensile test at constant temperature where failure occurs in a matter of minutes. If a sustained lighter load had been applied, a longer time would have been required for failure of the specimen. Therefore, if a succession of diminishing static breaking loads is applied to a group of specimens at a constant temperature, respective failures will take place at correspondingly increasing time intervals. A test of this nature is referred to as the stress-rupture type. The stresses at which rupture occurs are usually plotted as ordinates against the times for rupture as abscissae on log-log coordinates, and a linear relationship is generally found to exist for most alloys. In the majority of cases, tests of this nature are usually conducted for a period of about 1000 hr., and the linear relationship feature affords a ready means for extrapolation to longer time intervals. This latter operation should be pursued with caution, however, as erroneous predictions can result, especially if the material is susceptible to oxidation.

A conception of some representative stress-rupture values at 1200° F to 1500° F may be obtained from Fig. 10, where alloys 16-25-6 (3), S-590 (4), and 422-19 (5) have been selected at random for illustrative purposes. Amongst others, these three

TABLE I.

HIGH TEMPERATURE MATERIALS FOR GAS TURBINES.

Element	Material		
	16-25-6	S-590	422-19
C	0.06	0.49	0.45
Si	0.85	0.21	
Mn	1.54	0.60	
N	0.07		
Cr	16.55	19.50	28.00
Ni	25.20	19.78	16.00
Mo	6.46	3.95	6.00
Co		19.35	Bal.
W		4.15	
Cb		4.04	

materials show good high temperature properties and their compositions are given in Table I. Alloys 16-25-6 and S-590 are of the forged type, and 422-19 is precision cast. It can be seen that the cast alloy shows superior properties at high temperatures. The forged 16-25-6 alloy has been used in very large quantities for supercharger and jet propulsion gas turbine disks where temperatures are in the 1200° F. range. It has very good stress-rupture properties at such temperatures. For gas turbine disks operating in the vicinity of 1500° F, some of the more recently developed forged alloys, such as S-590 or S-816, show good characteristics. It will be noted that the relative difference between S-590 and 16-25-6 at 1200° F. is less than at 1500° F. For smaller parts, such as blading, which adapt themselves to the size limitations of the precision casting process, the cast alloys lend themselves to ready adoption and show good strength at high temperatures, as can be seen from the figure.

Reference to Fig. 10 will show that the higher the temperature, the more rapidly the material loses strength. Thus, the designer must take cognizance of the time factor involved when selecting or specifying materials. In steam turbine design where substantially lower temperatures are involved, the time element is not of as much significance as far as rate of loss of strength is concerned. In gas turbines, however, where elevated temperatures are necessary, the length of operating period must be considered.

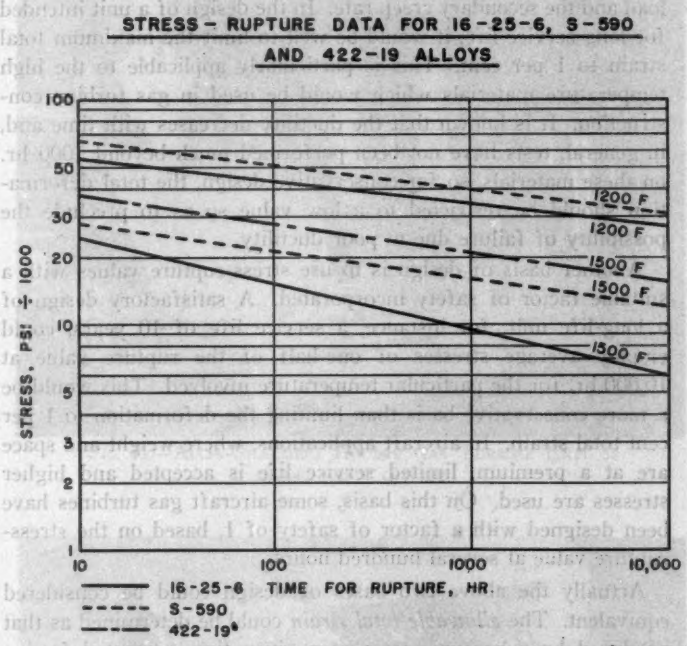


FIG. 10.—STRESS RUPTURE DATA FOR 16-25-6, S-590, AND 422-19 ALLOYS.

Therefore, in order to obtain a comprehensive representation of all of the factors involved in the design of a high temperature gas turbine, it is necessary to consider simultaneously: (1) the desired service life of the equipment, (2) the temperature at which it will operate, (3) the allowable deformation of the respective parts, and (4) the appropriate stress.

Knowing the time and temperature factors involved, a design could be based on the total allowable deformation of the respective elements of the equipment occurring in the time interval. The permissible stress would then be the one that resulted in the prescribed deformation in the time allotted and at the specified temperature for a given material. If the deformation is put on a relative basis, it is referred to as strain, and is stated in terms of per cent. It can be determined from the initial deformation under

load and the secondary creep rate. In the design of a unit intended for long service life, it would be well to limit the maximum total strain to 1 per cent. This is particularly applicable to the high temperature materials which would be used in gas turbine construction. It is known that the ductility decreases with time and, in general, tests have not been performed much beyond 1000 hr. on these materials, so for conservative design, the total deformation should be restricted to a low value so as to preclude the possibility of failure due to poor ductility.

Another basis of design is to use stress-rupture values with a suitable factor of safety incorporated. A satisfactory design of a long-life unit, for instance, a service life of 10 years, could employ average stresses of one-half of the rupture value at 10,000 hr. for the particular temperature involved. This would be a more conservative basis than limiting the deformation to 1 per cent total strain. In aircraft applications, where weight and space are at a premium, limited service life is accepted and higher stresses are used. On this basis, some aircraft gas turbines have been designed with a factor of safety of 1, based on the stress-rupture value at several hundred hours.

Actually the above two bases of design could be considered equivalent. The *allowable total strain* could be determined as that produced by a rupture stress at a given time corrected for an appropriate factor of safety. Conversely, an *allowable stress*, based on a rupture value incorporating a safety factor, could be selected so as to result in an acceptable total strain. In actual design practice, however, using the latter or rupture basis with a safety factor appears to be the most direct and readily applied procedure. Nevertheless, it should be remembered that both methods will produce the same result if they are mutually appropriately related.

From a design standpoint other properties such as creep, fatigue, ductility, damping capacity, oxidation, and corrosion, etc., should also be given consideration, but they are of secondary importance to stress-rupture.

Applications.

Various present day applications of the gas turbine, indicating the practicability of this type of equipment, have furthered its

FIG. 11.—ALLIS-CHALMERS HOUDRY GAS TURBINE UNIT OF 60,000 CFM CAPACITY.

FIG. 12.—ALLIS-CHALMERS 3500 HP., 1500 F., EXPERIMENTAL GAS TURBINE UNIT.

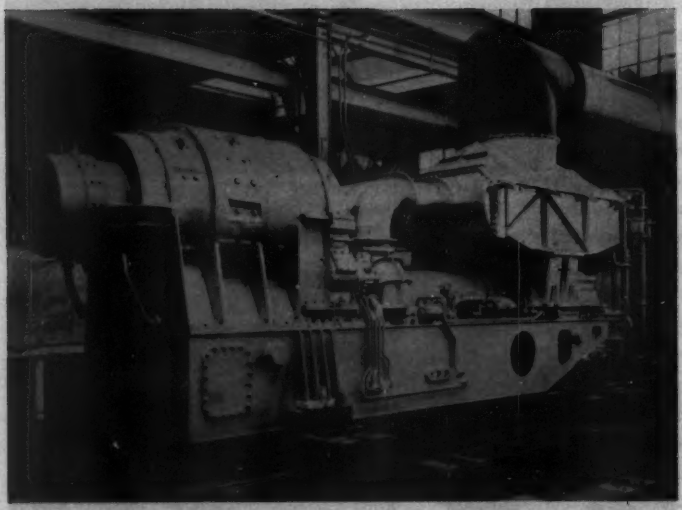


FIG. 13.—BROWN-BOVERI 2200 HP. LOCOMOTIVE GAS TURBINE UNIT.

emergence from a realm of academic interest to one of reality. The many advantages of the gas turbine make it a natural aspirant for numerous applications in land, marine, rail, and aircraft fields, and these will be dealt with individually below.

Land—The combustion gas turbine in the United States has had its principal commercial application in the oil refinery industry, where it operates in conjunction with the Houdry process (6). A typical installation view of such an application employing a 60,000 cfm. unit is shown in Fig. 11, wherein, from left to right, may be seen the gas turbine, axial compressor, reduction gear, generator, and a steam turbine for starting purposes. In this application, due to the pressure drop encountered in the process, the energy supplied to the generator is less than if the unit were used primarily for purposes of power production.

The first unit of this type has been in service since December, 1936. At present 28 of the Houdry gas turbines are in operation in this country. Of these, 6 were built by Brown-Boveri and 22 by Allis-Chalmers. The latter company has also built 10 additional units for installation in Europe.

A 4000 Kw. electric generating unit (7), built by Brown-Boveri, has been installed in an underground emergency standby power station in the City of Neuchatel, Switzerland. Since early in 1940 this unit has been operating for certain periods every week. Its simplicity and independence of water facilities ideally adapt the gas turbine to this class of service. Recently published tests indicate a coupling thermal efficiency of 18.04 per cent for this unit when operating on a non-regenerative cycle with a turbine inlet temperature of 1000 F. Regeneration was not adopted since it could not be economically justified because of the anticipated intermittent operation associated with this particular installation. This is the first unit designed for the primary purpose of producing power.

Brown-Boveri have also built a 10,000 Kw. gas turbine-electric generating unit intended as a standby set for the Filaret plant of Bucharest, Rumania. The machine consists of a two-shaft arrangement employing reheating and intercooling, but due to the standby nature of the service, no regenerator will be used. Tests on the unit showed a thermal efficiency of 23 per cent at full load, based on the lower heating value of the fuel. With a turbine inlet

temperature of 1112° F. and inlet air of 68° F., a maximum load of 12,000 Kw. was maintained.

Marine—The use of the gas turbine for marine service is one of the natural applications whereby its many advantages can be employed with benefit. The small space and weight characteristics of the gas turbine are particularly desirable for naval and maritime duty. The reversing operation when employing this type of unit aboard ship could be facilitated by the use of electric drive or a variable pitch propeller, thus eliminating the necessity of providing a separate astern turbine. Astern operation with gear drive is a more acute procedure in the case of the gas turbine since, unlike the steam turbine, it normally exhausts at atmospheric pressure, thereby requiring a separate vacuum system to provide a low density atmosphere in which to rotate the astern and ahead turbines during forward and reverse operation, respectively, so as to prevent prohibitive windage losses.

Prior to World War II, the Bureau of Ships of the U. S. Navy, recognizing the potentialities of the gas turbine, contracted with Allis-Chalmers to construct a plant designed for a temperature of 1500° F. and an output of 3500 Hp. This unit (8), the largest stationary gas turbine for this temperature in the world, is shown in Fig. 12 on the test floor of the U. S. Naval Engineering Experiment Station at Annapolis, Md. As of this date, it has successfully completed its tests at temperatures up to and including 1450° F., the highest operating temperature ever attained in a prime mover of this size. The next series of tests will be performed at the design temperature of 1500° F.

The cycle on which the unit operates consists of an axial-flow compressor driven by a turbine, while a separate turbine, arranged in parallel, furnishes the useful output. The air from the compressor enters the regenerator where it is preheated by the combined exhaust gas from both turbines. It next passes to two combustion chambers which furnish the high temperature motive fluid to the turbines.

The Elliott Company, under the sponsorship of the Bureau of Ships, manufactured a 2500 Hp. marine gas turbine plant (1). The unit is designed for a turbine temperature of 1200° F. A novel feature is the use of a Lysholm positive displacement type of compressor which has the advantage of a flat efficiency charac-

teristic, although its peak efficiency is not as high as that of the axial compressor.

The regenerative cycle with reheating and intercooling functions as follows: Atmospheric air enters the low-pressure compressor, from which it is discharged to the intercooler where its temperature is reduced. After next traversing the high-pressure compressor, it passes through the regenerator where it absorbs heat from the turbine exhaust gas. Fuel is burned with the heated air in a combustion chamber, and the resulting high temperature gas drives the high-pressure turbine which furnishes power to the low-pressure compressor. The exhaust gas from the high-pressure turbine, still containing adequate oxygen to support combustion, has its temperature increased by the burning of additional fuel in another combustion chamber. The ensuing products of combustion, at a temperature approximately the same as that at the inlet to the high-pressure turbine, propel the low-pressure turbine, which has the dual function of driving the low-pressure compressor and also furnishing the useful shaft output. The exhaust gas from the low-pressure turbine, after disposing of heat in the regenerator, is discharged to the atmosphere.

Based on the lower heating value of the fuel, tests on the plant showed coupling thermal efficiencies of approximately 29, 28, 26, and 20 per cent at full, ¾, ½, and ¼ load, respectively.

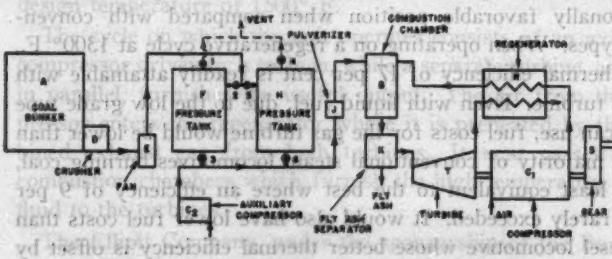
Rail—In all probability the most desirable application of the gas turbine is in the heavy traction field. When used as a locomotive drive, its many advantageous characteristics place it in an exceptionally favorable position when compared with conventional types. When operating on a regenerative cycle at 1300° F., a rail thermal efficiency of 17 per cent is readily attainable with the gas turbine. Even with liquid fuel, due to the low grade type that it can use, fuel costs for the gas turbine would be lower than for the majority of conventional steam locomotives burning coal, and at least equivalent to the best where an efficiency of 9 per cent is rarely exceeded. It would also have lower fuel costs than the Diesel locomotive whose better thermal efficiency is offset by the higher grade fuel which it uses. When coal fuel is used in the gas turbine cycle, the comparison becomes even more favorable.

In 1941 Brown-Boveri completed a 2200 Hp. gas turbine-electric locomotive (9) for the Swiss Federal Railways. A shop

view of the gas turbine power plant is shown in Fig. 13. From left to right may be seen the 200 Kw. auxiliary DC generator, main DC generator, reduction gear, regenerator, gas turbine, and vertical combustion chamber. The locomotive has a maximum specific service weight, including fuel, of 92 lb. per bhp. Its maximum speed is 70 mph. The tractive effort at starting is 29,000 lb. Acceptance tests of the power plant showed a maximum coupling thermal efficiency at ¾ load of 18 per cent based on the lower heating value of the fuel.

The Locomotive Development Committee (10) is now engaged in a research program which embraces the greatest potentialities regarding the application of the gas turbine as a railway power plant. This committee was formed as an agency of Bituminous Coal Research, Inc. in the latter part of 1944. The committee is composed of the chief executives of six leading coal-hauling railroads and three major coal companies. It was established for the purpose of developing a bituminous coal-burning locomotive which could compare favorably with the diesel locomotive. In the fulfillment of this aim the gas turbine has been selected as the prime mover which best adapts itself to the requirements involved.

The proposed arrangement of the equipment to be used on the coal-burning gas turbine locomotive is depicted schematically in Fig. 14. The coal is fed from bunker A, by means of a motor-operated screw, to crusher D where its size will be reduced to -16 mesh. Turbine exhaust gas will be used for drying the coal. By



SCHEMATIC CYCLE DIAGRAM FOR COAL-BURNING LOCOMOTIVE GAS TURBINE PROPOSED BY LOCOMOTIVE DEVELOPMENT COMMITTEE

FIG. 14.—CYCLE FOR COAL-BURNING LOCOMOTIVE GAS TURBINE PROPOSED BY LOCOMOTIVE DEVELOPMENT COMMITTEE.

means of fan E the coal is conveyed to pressure tanks F and H after it is crushed. The twin-tank system of coal storage is necessary due to the different pressure levels in the cycle. While one tank is being replenished, it is isolated by means of valves from the high pressure circuit, and coal is used from the other tank under full pressure.

For instance, when tank F was being filled, valves 1 and 2 would be open and 3, closed. The tank would then be under the low pressure existing in the conveying system. The crushed coal mixed with its conveying air enters the tank where an internal cyclone separates the coal from the air, the latter being vented. After tank F is filled and tank H exhausted, valves 1, 2, and 6 are closed and valves 3, 4, and 5 opened, whereupon the full pressure required for pulverizing exists in tank F.

The air compressor C_2 will produce a pressure of approximately 110 psig maximum required for pulverization. The intake will be connected with the discharge of the main compressor C_1, and a variable speed drive of C_2 will permit independent pressure relationships. After the crushed coal from tanks F or H is picked up by the high-pressure air, it is led to the inlet of the pulverizer J. This latter device was developed at the Institute of Gas Technology, of the Illinois Institute of Technology in Chicago, and was used to produce finely pulverized coal for gasification. It consists of a nozzle through which the compressed air, laden with crushed coal, is expanded to a pressure essentially equivalent to that at the turbine inlet, which will be approximately 60 psig at full load. Due to the sudden release of pressure occasioned by the rapid expansion, the air which had permeated the pores of the coal explodes, resulting in very complete pulverization. With proper design, a product as fine as 80 to 90 per cent of —325 mesh and 5 per cent or less of +100 mesh is obtainable with an air consumption of approximately one lb. per lb. of coal. Due to its extremely small size and effective pulverization, the adoption of this device appears to be the logical procedure for applications where space is at a premium, as in the case of locomotives.

The pulverized coal and primary air next enter combustion chamber B where they mix with the preheated air from the regenerator. The combustion chamber will comprise an inner shell, where burning will occur with about 40 per cent excess air, sur-

rounded by an annular space through which flows the diluent air. After mixture, the temperature of the gas entering the turbine will be 1300 F. Since the products leaving the combustion chamber contain fly ash, which would seriously erode the turbine blading, it is necessary to introduce an ash separator K. This device will consist of a series of Aerotec tubes which remove about 90 per cent of the entrained fly ash, operating on a cyclone principle.

The remaining equipment of Fig. 14 is the gas turbine power plant operating on a regenerative cycle of a type previously described. Since an electric transmission will be used on the locomotive, the gas turbine unit drives a direct current generator G through a reduction gear S. The full load thermal efficiency of the power plant will be 24 per cent referred to the input to the gear and based on the lower heating value of the fuel.

In fulfillment of the research program of the Locomotive Development Committee, two gas turbine power plants have been ordered and are in the process of construction. Both of the units operate on the cycle depicted in Fig. 14. One plant is being built by Allis-Chalmers and consists of a 6 stage reaction turbine driving a 20 stage axial flow compressor. Elliott is supplying the other which comprises a 4 stage reaction turbine driving a 2 stage centrifugal compressor. Locomotives with these power plants installed will probably be on the rails within a two-year period.

In 1939 Allis-Chalmers conducted an engineering study (11) of the combustion-gas turbine as a locomotive drive employing liquid fuels only. Consideration was given to four different gas-turbine locomotives including (a) a 5000 Hp. unit with hydromechanical transmission, driven by two separate power turbines, each receiving motive fluid from its main gas turbine-axial compressor set; (b) a 5000 Hp. unit with hydromechanical transmission, driven directly by two main gas turbine-axial compressor sets; (c) a 4500 Hp. unit with electrical transmission, driven directly by two main gas turbine-axial compressor-electric generator sets; and (d) a 2250 Hp. unit with electrical transmission, driven directly by a single main gas turbine-axial compressor-electric generator set. A heat exchanger was also incorporated in the latter design.

Fig. 15 shows a section through the 5000 Hp. locomotive with hydraulic transmission mentioned under (a) above. This locomo-

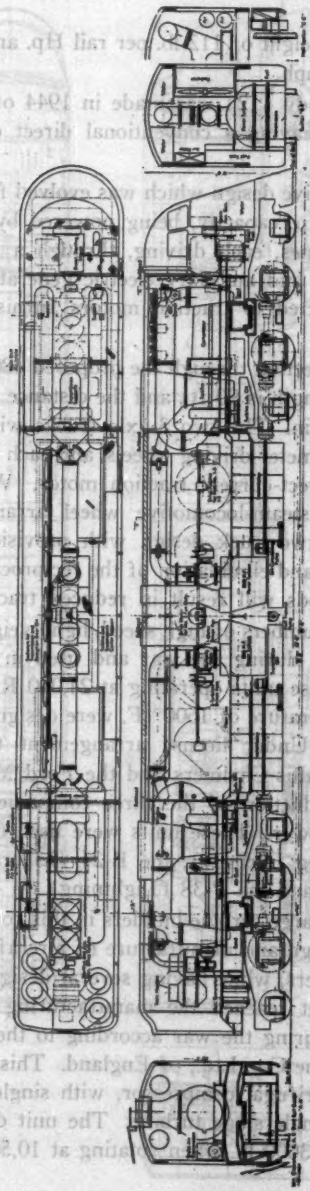


FIG. 15.—ARRANGEMENT OF ALLIS-CHALMERS PROPOSED 5000 HP. GAS TURBINE LOCOMOTIVE WITH HYDRO-MECHANICAL TRANSMISSION.

tive has a specific weight of 112 lb. per rail Hp. and is capable of speeds up to 120 mph.

An additional study (12) was made in 1944 of a gas turbine locomotive design having a conventional direct current electric transmission.

The road locomotive design which was evolved from the second study is of 4800 Hp. capacity, being powered by two duplicate 2400-Hp. gas turbines, each driving, through a reduction gear, direct-current main and auxiliary electric generators. The main generator is connected to traction motors furnishing power to the axles.

Fig. 16 shows a section through the proposed 4800-Hp. locomotive. The overall length is 90 ft., and the distance between bolster centers is 58 ft. 4 in. The two 3-axle trucks with 16-ft. wheel base have 52-in. diameter driving wheels, and each axle is powered by a nose-hung direct-current traction motor. When compared with conventional steam-locomotive wheel arrangements, it is expected that the two-truck design with provisions for proper lateral axle motion and elimination of the reciprocating motion of pistons and side rods will result in reduced track maintenance.

Aircraft—Vast numbers of high speed, light weight gas turbines were manufactured during the war and used in aircraft turbosuperchargers. These units operating at 24,000 Rpm. with a turbine inlet gas temperature of 1600° F. were designed and built by General Electric. Under license arrangement they were also manufactured by Allis-Chalmers and the Ford Motor Company. The combined production of the three companies totaled many millions of horsepower. These units were used on such aircraft as the B-17 (Flying Fortress), the B-24 (Liberator), the B-29 (Super Fortress), and the P-38 (Lightning).

All three of the large turbine builders in this country have been engaged in the design and manufacture of aircraft jet propulsion units. Allis-Chalmers, while having several designs of their own which have not yet reached the manufacturing stage, built jet propulsion units during the war according to the designs of the de Havilland Engine Co., Ltd., of England. This design employs a single stage centrifugal compressor, with single-sided entry of air, driven by a single stage turbine. The unit develops a static sea level thrust of 3000 lb. when rotating at 10,500 Rpm. with a

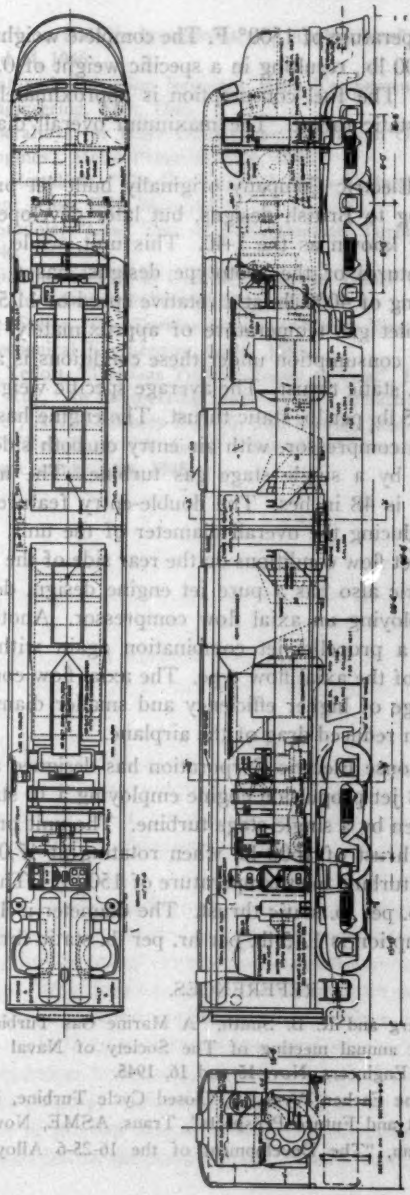


FIG. 16.—ARRANGEMENT OF ALLIS-CHALMERS PROPOSED 4800 HP. GAS TURBINE LOCOMOTIVE WITH ELECTRIC TRANSMISSION.

turbine inlet temperature of 1500° F. The complete weight with all auxiliaries is 1500 lb., resulting in a specific weight of 0.5 lb. per lb. static thrust. The fuel consumption is approximately 1.2 lb. per hr. per lb. static thrust. The maximum overall diameter is 50 inches.

The General Electric Company originally built jet propulsion engines according to British designs, but later developed a unit of higher rating known as the I-40. This unit, while retaining some of the features of the prototype designs, has a sea level static thrust rating of 4000 lb. at a rotative speed of 11,500 Rpm. and a turbine inlet gas temperature of approximately 1500° F. The specific fuel consumption under these conditions is about 1.2 lb. per hr. per lb. static thrust. The average specific weight of the unit is about 0.45 lb. per lb. static thrust. The engine has a single stage centrifugal compressor, with air entry on both sides of the impeller, driven by a single stage gas turbine. The maximum overall diameter is 48 inches. The double-entry feature has the advantage of reducing the overall diameter of the unit, although it complicates inlet flow conditions on the rear side of the impeller.

General Electric also has a pure jet engine design, designated as TG-180, employing an axial flow compressor. Another unit, the TG-100, is a propeller-jet combination again with a compressor element of the axial flow type. The axial flow compressor has the advantage of higher efficiency and smaller diameter, the latter resulting in reduced drag of the airplane.

The Westinghouse Electric Corporation has designed and built their X19XB-2B jet propulsion engine employing a 10 stage axial compressor driven by a single stage turbine. The unit produces a static sea level thrust of 1600 lb. when rotating at 17,000 Rpm. with a maximum turbine inlet temperature of 1500° F. The specific weight is 0.45 lb. per lb. static thrust. The diameter is 19 inches. The fuel consumption is 1.15 lb. per hr. per lb. static thrust.

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CONSIDERATIONS DETERMINING THE SIZE AND THICKNESS OF MAIN STEAM PIPING FOR NAVAL VESSELS.

By Rudolph Michel, Member.

Main steam piping systems for naval vessels must, in general, be fitted into more restricted spaces than is the case for central station power plants. The use of high steam temperatures will cause thermal expansion stresses to be high unless provision is made for absorbing the expansion in pipe bends or expansion joints. If in addition the system is designed for high steam pressures, relatively thick and inflexible pipes will result. In order to produce a piping system which represents the optimum in overall economy, the designing engineer must consider the following:

(a) Materials suitable for the steam temperature used.

(b) Allowable steam pressure and expansion stresses, as well as reaction forces.

(c) Allowable steam pressure drop through the piping.

(d) Erosion and noise caused by steam flow.

(e) Weight of the piping.

Considerable research has been done in the last decade on materials suitable for high steam temperatures, and it is not within the scope of this paper to discuss these in detail, but merely to enumerate the properties desirable in materials used for main steam piping service.

Main steam piping is occasionally subjected to vibrations excited by the propeller or some other source. In general, suitable means, such as the application of dampers, may be provided for reducing vibration amplitudes to negligible amounts. Shock loads affect mainly the stresses at the anchorage points, and the internal effects of transient vibrations excited by impact are neglected. Furthermore, the piping system is subjected to cyclic changes due to starting up and securing. During the average life of a destroyer, for example, this cycle will not be repeated more than 4000 times. If the piping is not subjected to thermal

shock, caused for example by rapid cooling due to a slug of water being carried over from the boiler, the stresses within the pipe result from steam pressure and expansion and are due to static loading. Fatigue failure, due to cyclic repetitions of load of the order of 1,000,000 or more cycles, have not been known to occur in main steam piping systems of naval vessels.

When a machine element is statically loaded at elevated temperatures, the properties of materials sought are high rupture strength, a high modulus of elasticity, good ductility and good creep resistance. The concepts of creep and rupture strengths are relatively new compared to those of the other properties mentioned.

The property of rupture strength, or stress required to cause rupture after a specified number of hours of loading at a given temperature is of great importance in high temperature piping design. Laboratory tests have shown that when a ferrous metal is subjected to a static load at elevated temperature, the material may fail suddenly, without necking down, if the stress exceeds a particular value which is dependent upon the temperature. The metal increases in size in the directions of the tensile stresses and for some applications this growth, or creep, must be kept within limits for purposes of maintaining minimum clearances. Rupture strength is spoken of as 500 hour, 10,000 hour, etc., rupture strength and decreases with increase in temperature as shown in Figure 1.

Creep stress is defined as the stress which causes a specified growth in a specified time at a particular temperature. Since in piping design we are in general not concerned whether a pipe increases slightly in diameter or length over a period of years, we are not primarily concerned with creep strength. There is a correlation between rupture and creep strength however, and materials having good rupture strength have a high resistance to creep. We thus have the phenomenon of a statically loaded material at elevated temperature changing its dimensions continually. This is in contrast to the same material subjected to the same stress at room temperature under which condition it returns to its original dimensions after removal of load provided the proportional limit has not been exceeded.

Figure 2 gives values of creep, rupture strength and yield strength of a steel which is well adapted for 1000°F. service. It should not be overlooked that ductility at operating tempera-

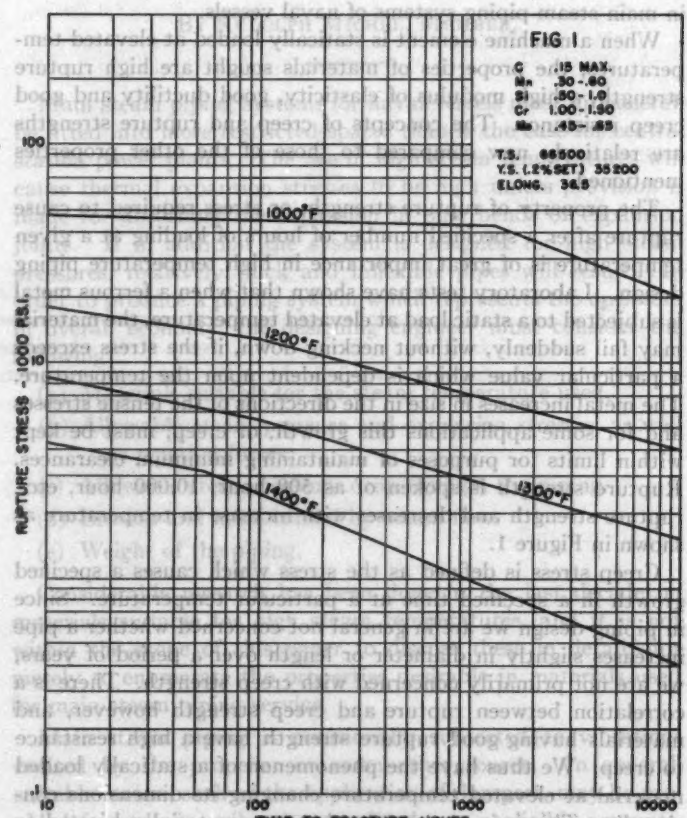
FIG 1
C .15 MAX.
Mn .30-.60
Si .50-1.0
Cr 1.00-1.50
Mo .45-.65
T.S. 66500
Y.S. (.2% SET) 35200
ELONG. 34.5
1000°F
1200°F
1300°F
1400°F
RUPTURE STRESS - 1000 P.S.I.
100
10
1
10
100
1000
10000
TIME TO FRACTURE - HOURS

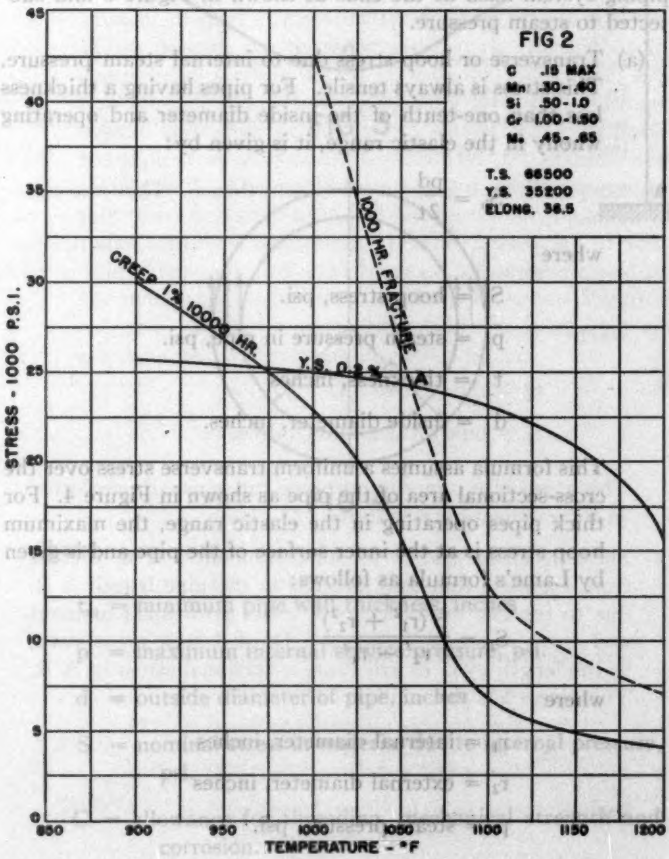
FIG 2
C .18 MAX
Mn .30-.60
Si .50-1.0
Cr 1.00-1.50
Mo .45-.85
T.S. 66500
Y.S. 35200
ELONG. 36.5
1000 HR. FRACTURE
CREEP 1% 10000 HR
Y.S. 0.2%
A
STRESS - 1000 P.S.I.
TEMPERATURE - °F
45
40
35
30
25
20
15
10
5
0
850
900
950
1000
1050
1100
1150
1200

ture is an important property of piping material, since it allows adjustments to take place within the metal which minimize the effects of high localized stresses.

The following principal stresses exist in a three-dimensional piping system fixed at the ends as shown in Figure 3 and subjected to steam pressure.

(a) Transverse or hoop stress due to internal steam pressure. This stress is always tensile. For pipes having a thickness less than one-tenth of the inside diameter and operating wholly in the elastic range, it is given by:

$$S_h = \frac{pd}{2t}$$

where

S_h = hoop stress, psi.

p = steam pressure in pipe, psi.

t = thickness, inches

d = inside diameter, inches.

This formula assumes a uniform transverse stress over the cross-sectional area of the pipe as shown in Figure 4. For thick pipes operating in the elastic range, the maximum hoop stress is at the inner surface of the pipe and is given by Lame's formula as follows:

$$S_h = \frac{p\,(r_1^2 + r_2^2)}{r_2^2 - r_1^2}$$

where

r_1 = internal diameter, inches

r_2 = external diameter, inches

p = steam pressure, psi.

The American Standards Association has adopted (1942) the following formula for determination of pipe thickness, in which the hoop stress appears as a nominal and not an actual value:

$$t_m = \frac{pd}{2S} + C$$

FIG 3

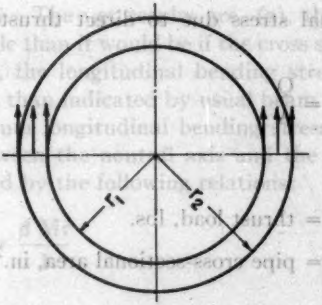

FIG 4

where

t_m = minimum pipe wall thickness, inches

p = maximum internal service pressure, psi.

d = outside diameter of pipe, inches

S = nominal stress in material due to internal pressure, psi.

C = allowance for threading, mechanical strength and corrosion.

(b) Longitudinal stress due to steam pressure as shown in Figure 5. This has a value equal to half the hoop stress,

$$S_{LP} = \frac{pd}{4t}$$

or

$$S_{LP} = \frac{p\,(r_1^2 + r_2^2)}{2\,(r_2^2 - r_1^2)}$$

It is always a tensile stress.

(c) Longitudinal stress due to pipe weight.
In a well supported pipe line, this stress is small and is usually neglected. Usual formulae for beam stresses (bending) may be employed.

(d) Longitudinal stress due to direct thrust. This has the value,

$$S = \frac{Q}{A}$$

where

Q = thrust load, lbs.

A = pipe cross-sectional area, in.^{2}

This stress is of minor importance and is usually neglected in three-dimensional systems.

(e) Shearing stress due to torsional moment. It is obvious from Figure 3 that any increase in the pipe length A B, due to thermal expansion, will cause a torsional moment and stress in the pipe length C D. Likewise an increase in the length of C D will cause a torsional stress in A B. This stress has the value:

$$S_s = \frac{32\,M_t\,r_2}{\pi\,(r_2^4 - r_1^4)}$$

where

S_s = shearing stress, psi.

M_t = torsional moment, inch pounds

r_2 = outer pipe radius, inches

r_1 = inner pipe radius, inches

(f) Longitudinal bending stress in a pipe bend. Consider the plain right angle pipe bend ACB of Figure 6, anchored at the ends and subjected to temperature stresses. After expansion, the pipe takes the shape ADB, and the pipe is subjected to reaction forces and bending moments as shown. Figure 7 shows an enlargement of portion D of Figure 6, with the tensile stresses at the outer fiber and compressive stresses at the concave or inner fiber. The resultants of the forces causing these stresses are directed toward the center of the pipe as shown, and thus flatten the pipe cross section as illustrated in the sectional view of Figure 7. The net results are, (a), the pipe becomes more flexible than it would be if the cross section remained round, (b), the longitudinal bending stress at the outer fiber is less than indicated by usual beam theory, and (c), the maximum longitudinal bending stress (see Figure 8) occurs between the neutral axis and the outer fiber and is calculated by the following relations:

$$S_{LB} = \frac{\beta M r}{I}$$

where

$$\beta = \frac{2}{3K}\sqrt{\frac{6h^2 + 5}{18}}$$

$$h = \frac{tR}{r^2}$$

S_{LB} = maximum value of longitudinal bending stress, psi.

M = bending moment, inch pounds

r = mean pipe radius, inches

I = moment of inertia of cross section of pipe before flattening, (inches)^4

$$K = \frac{1 + 12h^2}{10 + 12h^2}$$

t = pipe thickness, inches

R = radius of pipe bend at centerline, inches.

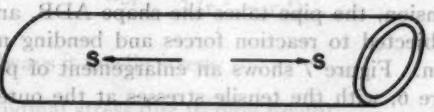

FIG 5

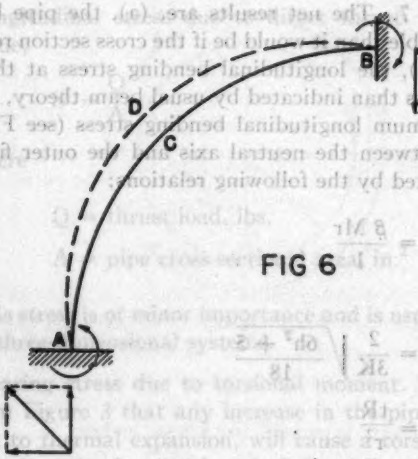

FIG 6

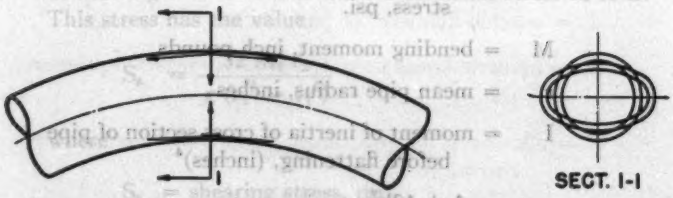

FIG 7

The flattening of the pipe at a bend also causes transverse bending stresses which vary over the cross section as shown in Figure 9. The transverse bending stress is fortunately highly localized and does not extend over a large length of the pipe bend. In systems designed for not over 600 psi. pressure and 850°F. these stresses have been ignored when computing total combined stress. As pressures are increased and pipes become thicker, the transverse bending stress may reach values so high that these stresses exceed the rupture strength of the material even after considerable plastic deformation and yielding has taken place. Transverse bending stress should therefore be carefully considered when temperatures and pressures of high order of magnitude are used. A material with good ductility and relaxation properties at the operating temperature is particularly desirable. Transverse bending stress is given by:

$$S_{TB} = \frac{18\ M r_o h \left(1 - \frac{2y^2}{r_o^2}\right)}{I\ (1 + 12h^2)}$$

where

M = bending moment, inch pounds

r_o = outer radius, inches

$h = \frac{tR}{r_o^2}$

R = mean radius of bend, inches

I = moment of inertia, inches^4

y = 0 or r_o for maximum value of S_{TB}

Graphical, analytical and experimental methods are in use for determining the moments acting on a piping system. For a pipe which lies wholly in one plane, analytical methods are relatively simple. For three dimensional systems, analytical and graphical methods become laborious and certain agencies are equipped to make models of the system. After impressing the computed displacements, due to thermal expansion, on the system, the reaction forces are measured on spring balances and the moments may then be computed easily. The reader is referred to the bibliography for methods of determining bending and torsional moments.

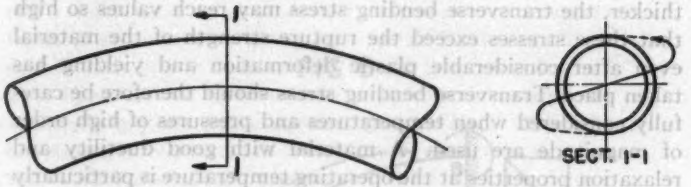

FIG 8

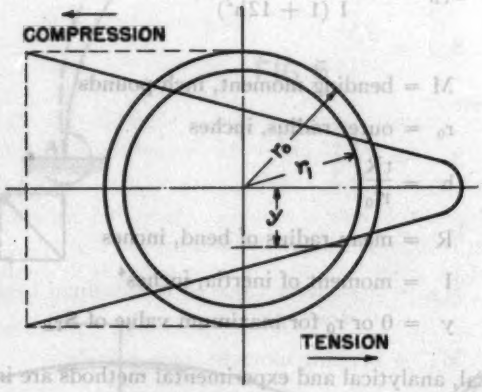

FIG 9

From the above considerations it is apparent that for a particular layout, the stresses occurring in the piping system are dependent largely upon pipe size and thickness, steam pressure, and expansion. Longitudinal and transverse bending stresses are both affected by pipe thickness. The pipe thickness must be such that the total combined stress will not exceed allowable limits. The A.S.M.E. Piping Code and the American Standards Association have set up values of hoop stress to be used for various materials at different temperatures. These are adequate for the lower temperature range, but for values of temperature at or near 1000°F. result in pipe thicknesses that could not be tolerated under the restricted conditions prevailing on naval vessels. It will be found that it is not desirable to select a material for operating temperatures in the range 850°–1000°F. which will necessitate the use of hoop stresses below 5000 psi. and in general higher hoop stresses are desirable.

Regarding a criterion of failure, we are guided to some extent by considerations similar to those prevailing in other statically loaded structures. For example, a structural beam is not considered to have failed when highly localized stresses exceed the yield point, since the beam as a whole still retains its elasticity. In like manner, we seek a method of combining the component stresses that will give a combined stress which is indicative of the yielding of the pipe as a whole at the operating temperature. Two theories of failure are applicable, viz., the Henckey-von Mises or strain-energy theory and the maximum shear theory (Coulomb-Guest).

If failure is associated with yielding, the strain energy theory applies in accordance with the following formula:

$$S = \sqrt{3\,S_s^2 - S_L\,S_h + 2\,S_L^2}$$

where

S = combined stress indicative of yielding of pipe as a whole, psi.

S_s = shear stress due to torsion, psi.

S_L = sum of longitudinal pressure and bending stresses, psi.

S_h = hoop stress, psi.

This formula predicts yielding of the whole pipe more accurately than that based on the maximum shear theory.

If the pipe is given 50 per cent cold springing, the computed combined stress will not be attained in either the hot or cold condition, even though the system is designed to be stressed up to the yield point. Cold springing therefore introduces a factor of safety for the maximum stress attained by introducing bending stresses of reversed sign when cold.

If the rupture strength of the material at operating temperature for the accepted life is less than the yield strength, it will of course fail by rupture before the pipe as a whole yields. If yielding of the pipe as a whole is taken as the criterion of failure, the maximum usable temperature is indicated by the point at which the rupture strength and yield strength curves intersect, as at point A, Figure 2. If it is desired to utilize a given material above this temperature, rupture strength will be the criterion of failure and the maximum shear theory advanced by Bailey is recommended for computing combined stress. This theory can of course be applied for all ranges of temperature and is recommended where particularly thin pipe walls are desired. The reader is referred to the bibliography for a detailed description of Bailey's theory.

Pressure drop in a steam pipe is due to friction of the steam against the pipe walls and between molecules. In cases where the total drop is not in excess of ten per cent of the initial pressure, the following formula holds:

$$\Delta P = \frac{f\ L\ V^2}{772\ d v}$$

where

ΔP = pressure drop, psi.

f = friction factor

L = equivalent length of the piping system, feet

V = velocity of steam, feet per second

d = inside diameter of pipe, inches

v = specific volume, cu. ft. per pound.

The friction factor is dependent upon the condition of the pipe walls, that is, whether smooth and new or corroded, scaled or otherwise roughened. Friction factor is furthermore a function of Reynold's number and in general decreases with an increase of Reynold's number up to about 5,000,000. Beyond this point the friction factor for superheated steam flow in new piping is essentially a constant and its value is closely approximated by the dimensionless quality 0.015.

The equivalent length of the system consists of the length of straight piping plus the lengths of the valves, bends and fittings, expressed in terms of straight piping. It has been ascertained by experiments made at the U. S. Naval Boiler and Turbine Laboratory that for the flow of steam, which is usually at high Reynold's numbers, the equivalent length of the pipe system is essentially that of the equivalent length of the sum of the various component parts. Thus, for the average arrangement of a naval main steam piping system, the grouping of valves and fittings does not affect the equivalent length of the system.

Steam velocity is computed by means of the relation:

$$V = 3.056 \frac{Wv}{d^2}$$

where

V = steam velocity, feet per second

W = steam flow, lbs. per minute

v = specific volume, cu. ft. per pound

d = inside pipe diameter, inches.

Factors affecting the velocity of the steam at full power are the allowable pressure drop, noise and erosion of the pipe, valves or fittings. It should be remembered that most naval vessels cruise at speeds corresponding to about 10 to 20 per cent of full power for a large portion of their operation, so that greater pressure drops can be tolerated at full power than in the case of central stations. Erosion of piping, valves and fittings is therefore negligible, as is the noise at cruising speeds caused by the steam passing through restricted openings. Serious erosion will occur only if moisture is present in quantities greater than ten per cent for steam having velocities over 500 feet per second.

In main steam lines the steam is without moisture content, being superheated, and the velocities are in general of the order of one-fourth of those existing in the last turbine stages.

Noise may become excessive if steam velocities exceed about 400 ft. per second. Very high velocities are premissible in piping used only occasionally, such as astern steam lines and cross-over pipe lines connecting forward and after machinery spaces.

Aside from considerations of stresses, it is apparent that the choice of main steam piping sizes is dictated mainly by considerations of allowable pressure drop at full power operation. Friction pressure drop is a throttling process and represents a loss in available energy in the turbine. It involves an increase of entropy in the steam as it flows along the pipeline. The loss in available energy may be computed from the relation:

$$U.E. = \Delta S \times T$$

where

U. E. = unavailable energy, Btu. per lb.

ΔS = change in entropy between initial and final steam conditions in the pipe line.

T = absolute temperature corresponding to condenser pressure (sink temperature).

Since the horsepower output of a steam turbine is essentially proportional to the isentropic heat drop in the machine and the pounds of steam passed, we can compute the additional steam required by the turbine due to the decrease in available energy per pound. The following example will illustrate the point: Assume a destroyer power plant designed for 800 psi. g. 900°F. at the superheater outlet and a 27½" hg vacuum in the condenser at full power. Two main turbines, of 30,000 Hp. each, have a basic steam rate of 6 lbs. per hp.-hour at full power. The piping is of carbon-molybdenum steel. Two boilers supply steam to each turbine at full power. It is desired to estimate the difference in fuel oil consumption when 6-inch I.P.S. and 5-inch I.P.S. piping is used between the superheater outlet and the Y connection ahead of the turbine throttle.

Consider first the 6-inch pipe size. The total equivalent length of piping, valves, fittings and bends is estimated to be 400 feet.

Using the American Standards Assoc. formula for pipe thickness, we get,

$$t = \frac{pd}{2S} + 0.065 = \frac{800 \times 6.625}{2 \times 10{,}000} + .065 = 0.33 \text{ in.}$$

The inside pipe diameter is, $6.625 - 2 \times 0.33 = 5.965$ inches. Since the steam flow is 1500 lbs. per minute, the velocity is,

$$V = 3.056 \frac{Wv}{d^2} = 3.056 \times 1500 \times \frac{0.951}{(5.965)^2} = 122 \text{ ft. per sec.}$$

The pressure drop to the Y-connection will be,

$$\Delta P = \frac{f\,L\,V^2}{772\,dv} = \frac{0.015 \times 400 \times (122)^2}{772 \times 5.965 \times 0.951} = 20.4 \text{ psi.}$$

The loss in available energy is computed as follows, assuming a 5°F. drop in temperature of the steam:

Entropy at 795 psi. a.	— 895°F.	= 1.641 Btu. per °R.
Entropy at 815 psi. a.	— 900°F.	= 1.638 Btu. per °R.
Increase in entropy		= 0.003 Btu. per °R.

Absolute temperature corresponding to

$27\frac{1}{2}$" hg. = 108.7 + 460 = 568.7° F.A.

Loss in availability = $0.003 \times 568.7 = 1.7$ Btu. per lb. steam. This represents a fuel loss of

$$\frac{1.7 \times 6 \times 60{,}000}{18{,}500 \times 0.83} = 40$$

lbs. oil per hour, assuming the boiler efficiency to be 83 per cent.

If a 5-inch pipe is used the equivalent length will be approximately 360 feet. The calculations for pressure drop and loss in available energy follow.

$$t = \frac{800 \times 5.625}{2 \times 10{,}000} + 0.065 = 0.29 \text{ inches pipe thickness.}$$

Inside pipe diameter = $5.625 - 2 \times .29 = 5.045$ inches.

$$\text{Steam velocity} = 3.056 \times 15.00 \times \frac{0.951}{(5.045)^2} = 172 \text{ ft. per sec.}$$

Pressure drop to Y connection,

$$\Delta P = \frac{0.015 \times 360(172)^2}{772 \times 5.045 \times 0.951} = 43 \text{ psi.}$$

Entropy at 772 psi. a. — 895°F. = 1.644 Btu. per °R.

Entropy at 815 psi. a. — 900°F. = 1.638 Btu. per °R.

Change in entropy = 0.006 Btu. per °R.

Loss in available energy = $0.006 \times 568.7 = 3.4$ Btu. per lb. This represents a fuel loss of

$$\frac{3.4 \times 6 \times 60{,}000}{18500 \times .83} = 80$$

lbs. per hour.

The 6-inch pipe therefore has a thermodynamic advantage over the 5-inch pipe, represented by a gain in available heat drop in the turbine, equivalent to 80 — 40 = 40 lbs. of oil consumed per hour.

Since radius of action at cruising speeds is the prime consideration in the design of most naval vessels, the additional oil weight represented by the loss in economy should be balanced against the decreased weight of piping. Naturally, if pipe size is excessively reduced, the turbine size and weight must be increased due to meeting full power requirements, so that there exists an optimum pipe size for each installation. The smaller pipes have a distinct advantage over larger ones in that less thrust exists at the anchorage points and bending stresses due to expansion are reduced.

To summarize, it is desirable in naval main steam piping systems to select materials and pipe sizes that will allow the use of minimum pipe wall thicknesses. This is especially the case when a reliable expansion joint is not available and pipe bends are used to obtain flexibility. The smaller and thinner pipes possess, in addition to greater flexibility, the advantage of lighter weight and smaller thrust loads at the fixed points. The decrease in pipe size will be accompanied by increased pressure drop in the steam. The optimum pipe size is reached when the increase in cruising radius, due to weight saved by the use of smaller pipe sizes, is balanced by the decrease in cruising radius, due to poorer turbine economy and the resultant increase in machinery

and fuel weight. Rupture strength is recommended as a criterion of failure where the life of the piping will be limited to a known number of hours.

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OFFICIAL ABSTRACT
FINAL REPORT
OF THE
BOARD TO INVESTIGATE THE DESIGN AND METHODS OF CONSTRUCTION OF WELDED STEEL MERCHANT VESSELS.

Early in the war certain welded merchant vessels experienced structural difficulties in the form of fractures which were not readily explained. These fractures in many cases manifested themselves with explosive suddenness and exhibited a quality of brittleness which was not ordinarily associated with the behavior of a normally ductile material such as ship steel.

The seriousness of this structural failure problem may be gaged by the fact that out of approximately 5000 ships constructed in the course of the war, about one-fifth sustained casualties. Of these casualties, 127 were classified as serious. Eight vessels were lost, but it is extremely fortunate that only 26 persons lost their lives. Contrary to widespread belief, the fracture menace did not confine itself to Liberty Ships but involved vessels of all types.

Since the fractures sustained occurred under normal operating conditions and were not the result of battle damage, it was evident that their implications might have been far reaching and have had a signal effect upon the war effort. Accordingly, in April 1943, a Board was appointed by the Secretary of the Navy to investigate the circumstances surrounding the structural failures reported. This Board was formally known as the Board to Investigate the Design and Methods of Construction of Welded Steel Merchant Vessels, and consisted of: The Engineer-in-Chief, U. S. Coast Guard as Chairman; and the Chief of the Bureau of Ships, U. S. Navy; the Vice Chairman, U. S. Maritime Commission and the Vice President-Chief Surveyor, American Bureau of Shipping as members.

The investigation of this Board has been in progress for more than three years. During that time two interim reports were made to the Secretary of the Navy. A Final Report covering the Board's activities has been submitted to and approved by the Secretary of the Navy and the following facts are abstracted from that Report.

During the course of the inquiry, many research projects were initiated under the National Defense Research Committee. These projects were carried out with the services of the Welding Group of the War Metallurgy Committee at the National Research Council. In utilizing the facilities of the War Metallurgy Committee, the Board was able to bring to bear on its problems some of the best scientific minds in the country. Through a complete coordination of effort in research, results were obtained in a minimum of time. Close liaison was maintained throughout the investigation with the British and with other Allied Nations who were concerned with the operation of welded merchant vessels.

The over-all designs of merchant vessels were checked by recalculation of the longitudinal strength and by means of static structural tests on certain vessels. It was found that there was a margin of strength in every case over that required by existing standards and that the basic analytical method used in calculating the strength of the hull girder is valid.

In the investigation of detail design, it became apparent that the monolithic character of the welded ship resulting from the method of fabrication can produce high stress concentrations and severe restraint, thereby tending to inhibit plastic flow. This condition did not exist generally in the riveted ship. The danger of high concentration at points of structural discontinuities in the welded ship is further aggravated by welding usually present at such points. Every fracture examined started in a geometric discontinuity or notch resulting from unsuitable design or poor workmanship.

The investigation pertaining to structural details has strongly emphasized that too much attention cannot be paid to the elimination of discontinuities

or notches, whether they be small or large, and that the effect of discontinuities is aggravated by welding.

Studies indicated that steel as furnished to shipyards complied in every respect with present physical requirements. In spite of this, impact tests of steel samples taken from vessels which had suffered fractures indicated that in many cases the steel was notch sensitive. In addition, it was found that some steels furnished to shipyards were also notch sensitive. There is a necessity for the establishment of a new specification to include a practical test for the evaluation of the notch sensitivity of commercial steels.

There is no indication that inferior quality or misapplication of welding electrodes was responsible for welded ship fractures. This does not mean, however, that an improvement in electrodes and covering materials might not be beneficial.

The use of welding in the construction of merchant vessels during the war permitted the launching of an enormous fleet of ships which played a vital part in the winning of the war. There are, however, certain disadvantages connected with welding which were not fully realized at the outset. Although the technique of depositing weld metal and the application of welding sequences to minimize shrinkage, distortion and cracking were fairly well understood, relatively little was known of deleterious conditions accompanying the welding processes on large ship structures. Consequently, when factures in all-welded steel merchant vessels first began to manifest themselves (as in the *Schenectady* and the *Esso Manhattan*), conditions were found which did not conform to previous experience. There was a general feeling that the accelerated shipbuilding program and the concomitant quantity production of all-welded ships had resulted in a general disregard for proper construction procedures and workmanship. It was particularly felt that insufficient care was being devoted to welding sequences with the result that locked-in stresses were present in many ships to a higher degree than would be expected. The presence of these higher stresses was believed to be an important factor in the incidence of the observed fractures. The results of the investigation, however, have not substantiated this belief.

Although a large amount of work was conducted in the investigation of residual and locked-in stresses resulting in a considerable extension of knowledge in this respect, no evidence has been found to indicate that these stresses are important in causing the factures of welded ships.

The feeling that workmanship had suffered due to the pressure of wartime production programs was substantiated. The importance of maintaining adequate standards of workmanship has been clearly established by the analysis of structural failures in the past three years. Poor workmanship engenders fractures since a fracture may originate at a small notch such as occasioned by peened-over cracks and under-cut welds, by porosity and inclusions in the welds, or by "saddle" welds resulting from incomplete penetration, which leaves voids in the center of the joints. High quality workmanship is still an important need in the building of welded ships.

A study of operating conditions resulted in the finding that loading and ballasting procedures did not create abnormal bending moments.

The war time operation of cargo ships in convoys and overseas routes which are only infrequently used in normal times imposed unusual hardships on the vessels, especially during the early part of the war when convoys were being routed through extremely cold waters where heavy seas prevailed during the winter months. The highest incidence of fractures occurred under the combination of low temperatures and heavy seas. The risks involved were accepted as far as heavy seas were concerned, but at the start the adverse effects of low temperature were not fully appreciated. When these facts were recognized, vessels modified to increase their resistance to fracture were assigned to the more rigorous trade routes.

The Board concluded that the fractures in welded ships were caused by notches built into the vessels, either through design or as the result of workmanship practices, and by steel which was notch sensitive at operating tem-

peratures. When an adverse combination of these factors occurs, the ship may be unable to resist the bending moments of normal service.

The serious epidemic of fractures in the steel structures of welded merchant vessels has been curbed through the combined effect of the corrective measures taken on the structure of ships during construction and after completion, improvements in design, and improved construction practices in shipyards.

The results of the investigation have vindicated the all-welded ship. Statistics show that the percentage of ships sustaining serious fracture is small. With proper detail design, high quality workmanship and a steel which has low notch sensitivity at operating temperatures, a satisfactory all-welded ship structure may be obtained.

However, until experience can be had with vessels constructed under normal conditions of improved design, materials, and workmanship, some form of crack arrestor, such as a riveted gunwale angle, should be incorporated in the hull girder of all large welded vessels.

The research program conducted in connection with the investigation has produced at least partial answers to most of the more urgent questions and has given an adequate solution for the purposes of the Board. It now becomes necessary to assure the continuance and the extension of this work. Paths have been indicated along which real improvement can be made in structural designs, materials and fabrication methods. The Board has been dissolved—it cannot follow these leads—however it is important that we maintain our present position in maritime affairs and protect our standing in world wide competition by continuing fundamental research work on design and methods of construction of steel ships. Accordingly, the Secretary of the Treasury has appointed a Committee to carry on the work of the Board. It has been designated as the Ship Structure Committee. This Committee is jointly controlled and financed by the agencies which comprised the Board, namely, the Bureau of Ships, U. S. Navy; the U. S. Coast Guard; the U. S. Maritime Commission; and the American Bureau of Shipping.

FUNDAMENTALS OF SHIP STABILITY.

This article by J. S. Redshaw is reprinted from the Transactions of the Institute of Marine Engineers for 1947.

Synopsis.—This paper gives firstly an interpretation of fundamentals and an historical note. It then deals with transverse stability; longitudinal stability; motion of ships in a sea-way; center of gravity; inclining experiments (tanometer angle recorder); transverse stability demonstration model; free surface effects; flooding and damage control demonstration model; anti-rolling devices including bilge keels, anti-rolling tanks, gyro stabilizers, and actuated fins; stability of submarines; and standards of stability (American proposals, etc.).

Recently two very comprehensive *papers dealing with the naval architect's contribution to the science and art of ship propulsion, have been presented to the members of this Institute.

The first of these was concerned with the geometric characteristics or "form" of the hull of a mechanically-propelled ship in its bearing on the economics of power propulsion, and the author has no doubt that the contents of both papers effectively reassured the members that their companion societies and technicians are devoting adequate effort to ensuring the proper and efficient application of the mechanical power placed at their disposal by marine engineers.

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* "Fundamentals of Ship Form", by F. H. Todd, B.Sc., Ph.D. Transactions of The Institute of Marine Engineers, Vol. LVII, No. 1, February, 1945, pp. 1-29.
"Fundamentals of Ship Propulsion", by F. H. Todd, B.Sc., Ph.D. Transaction of The Institute of Marine Engineers, Vol. LVIII, No. 2, March, 1946, pp. 23-38.

One ventures the opinion that your Council has chosen an appropriate sequel to complete the series by directing attention to another fundamental characteristic related specifically to the form and proportions of a ship's hull —the very important subject of stability—so that marine engineers may have the further assurance that the safety and behavior of the container, to which they entrust their complicated and expensive apparatus with its expert operating personnel, receives proper study and attention.

A ship is nothing more or less than a transport vehicle and, being a co-operative product of the naval architect and marine engineer, many of the author's introductory remarks on the aforesaid papers on form and propulsion apply with equal force to stability.

For in this matter of stability and ship behavior, the engineer contributes in most cases a substantial proportion of the weight of the finished article, and the size and location of the propelling plant must influence, in a very real sense, those elements of form and dimensions for which the naval architect assumes responsibility. Hence the desirability of engendering a general understanding of one another's problems, and the need for closer partnership in seeking their satisfactory solutions.

The systematic acquisition of knowledge concerning the propulsion aspect of ships' forms is of relatively recent date, beginning virtually with the practice of methodical model experiments, and though a large field has been covered, continued progress in research and development is still the order of the day.

On the other hand, the knowledge of the mechanics of stability of form of floating vessels has a considerably older background and the long dynasty of investigators and theorists has left little, if any, unknown or unexplored territory. The mathematical treatment of the subject is dated by the fact that the mystic word "metacenter" first appeared in the technical language of our profession exactly 200 years ago, so it will be appreciated that as the principles of geometry and mathematics have not sensibly changed in the interval, the subject is by this time in itself well "stablized".

To recount the fundamental principles imposes no greater responsibility on a writer than to review and condense into a concise pattern the field of knowledge now covered—and it is the purport of this paper to make, if possible, a picture to be viewed in passing by those whose more intimate interests and activities lie in other directions.

Dipping very briefly into the history of the study of stability of floating vessels, there seems to have been little activity worth recording before the seventeenth century; but one finds in the early works on naval architecture published in this country, references to the extensive mathematical explorations by foreign investigators, French, Russians, Swedes, etc., from roughly the end of the seventeenth century, onwards. Scott-Russell in his treatise on naval architecture of 1865 writes feelingly of the woeful lag before this foreign enterprise made any impression on the British builders, who for so long had viewed the intrusion of the mathematician or scientific worker into the field of shipbuilding with a marked lack of enthusiasm, to put it mildly. The story of that period does not make very happy reading and is in strange contrast to the pioneering and progressive spirit exhibited by the British technicians in the field of ship resistance and propulsion.

However, when a start was made, much was written on stability as contributions to learned societies or as part of more comprehensive works such as that of Scott-Russell already mentioned, and Professor Macquorn Rankine's treatise on theoretical and practical shipbuilding published in 1866.

Some 20 years later there appeared what was claimed to be the first British publication devoted entirely to stability. This was Sir Edward Reed's treatise on the stability of ships in 1885, an exceedingly detailed and extensive work, which contained a survey of many of the foreign contributions to the subject up to that time. Reed's stability had, and still retains, a wide vogue as a standard text-book, and was later followed by the familiar works of Attwood, Biles and others, each containing sections dealing, in more abbreviated form, with the now generally accepted principles of stability.

The word "fundamentals" in the title of this paper may admit of a variety of interpretations according to the point of view. To the practical operator of a ship the simple fundamental concern is that as it is intended to float and navigate in a substantially upright attitude, and the ship must be so designed that any deviation, through whatever cause, must automatically generate a resisting force to restore the situation. The more scientific approach is concerned with study and analysis of the characteristics which distinguish a stable from an unstable condition, and by which the foregoing state of affairs is assured. This resolves into a conflict or balance between gravitational forces on the ship coupled with any applied forces on the one hand, and the opposing or buoyant forces from the water on the other hand. As the amount and direction of the latter are determined entirely by the geometric properties of the form of the ship's hull, it will be appropriate to begin with brief definitions of the conventional nomenclature used in this connection. Though some of the terms are not related exclusively to stability they are included here as a general reference.

The orthodox ship form is bounded by curvilinear surfaces and has only one plane of symmetry—a vertical plane through the longitudinal center line. The character of the curved surfaces is defined in what is called a *lines plan* which shows in three projections the traces or outlines that would be produced if the whole body were cut by a series of horizontal and vertical planes.

The traces produced by the horizontal slicing are called *water-planes*, and in the other projections there are the *transverse sections* across the ship, and the *bow lines* and *buttocks* on the longitudinal direction.

Displacement is the weight in tons of the water that would fill the space occupied by the immersed part of the ship and is of course exactly equal to the total weight of the ship as it floats. Usually displacement expressed in tons weight is denoted by symbol Δ and to convey the equivalent volume in cubic feet the symbol is reversed ∇.

To indicate the shape or characteristics of this immersed form or displacement, it is customary to use a series of coefficients which have the following significance:—

Block Coefficient expresses the ratio or proportion that the actual volume of displacement bears to the volume of an imaginary containing rectangular box having the same overall length, breadth and depth. Referring to Fig. 1, a block coefficient of say 0.50 would mean that the volume of displacement is just one half of the volume of the rectangular box $AC \times AB \times AE$ and the coefficient, therefore, gives a picture of the general "bulkiness" of the form.

Obviously with the same block coefficient the displacement can be distributed in a variety of ways along the length, so to convey a notion of a particular distribution we resort to the:—

Prismatic or Longitudinal Coefficient which expresses the ratio or propotion that the actual volume of displacement bears to the volume of an imaginary container having the same overall length and a constant section similar in shape to the largest middle section of the ship form. This is illustrated in Fig. 2, and it is evident that the prismatic coefficient conveys the idea of distribution of displacement as between the middle portion and the ends.

Midship Area Coefficient which expresses the ratio between the largest cross section and its circumscribing rectangle (*GHKJ* in Fig. 1) is the link between the block and prismatic coefficient, for it is obvious that with a given block coefficient, a fine or small midship coefficient results in a full or large prismatic coefficient, which means that the displacement is spread more towards the ends and indicates what is termed a "full-lined" ship.

Waterplane Coefficient expresses the ratio between the area of the horizontal waterplane and its circumscribing rectangle (*AEFC* in Fig. 1); and the centroid of the waterplane area is called the *Center of Flotation* (*C.F.*).

Vertical Prismatic Coefficient expresses the ratio between the volume of displacement and an imaginary container having the same draft as the ship and a constant horizontal section similar to the load waterplane. This coefficient

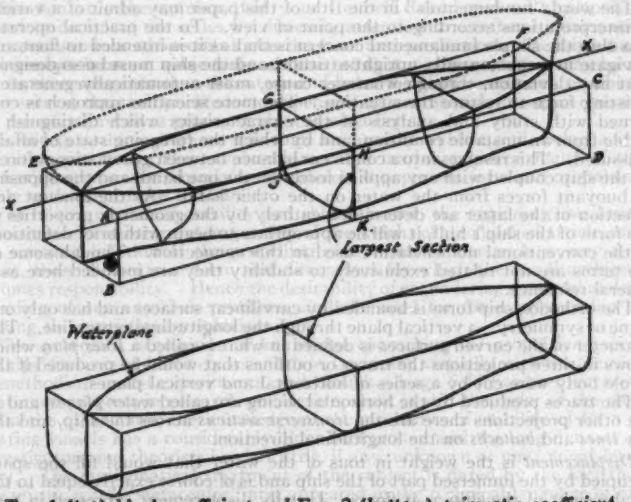


FIG. 1 (*top*) *block coefficient and* FIG. 2 (*bottom*) *prismatic coefficient.*

conveys the idea of the vertical distribution of displacement as between the upper and lower levels and therefore has a particular bearing on stability. A small or "fine" vertical coefficient indicates that the bulk of the volume is bunched up towards the higher levels which as will be seen later tends towards greater stability.

Center of Buoyancy (*B*) is the geometric centroid of the volume of displacement. All the buoyant forces by which the water supports the ship can be integrated into a single vertical force acting through the center of buoyancy, in just the same way that all the component weights of the ship and its contents can be integrated into a single vertical force or weight acting through its *Center of Gravity* (*G*).

Righting Arm. If the opposing forces of gravity and buoyancy, acting downwards through the center of gravity, and upwards through the center of buoyancy respectively, are not in the same vertical plane, a couple is created tending to move the ship either towards or away from the upright position. The arm of this couple is called the "righting arm" or lever (generally denoted by *GZ*) and if the tendency of the couple is towards the upright, the ship is said to be *stable* and if away from the upright, *unstable*.

Transverse Stability. For an orthodox ship form the breadth is only a fraction of the length, and as the critical stability condition relates to the shorter dimension, primary importance attaches to the athwartship or transverse stability.

In a first approach to the question of stability, an end-on view of, say, a passenger liner floating serenely upright with high superstructures and top hamper obviously bringing its center of gravity well above the apparently "soft" support from the water, seems to conflict with one's conception of stability in the mechanical sense. It looks like a mobile pendulous system on which we are accustomed to find stability contingent upon the center of gravity being below the point of support! and this apparent anomaly probably gives rise to some confusion in the mind of the casual observer.

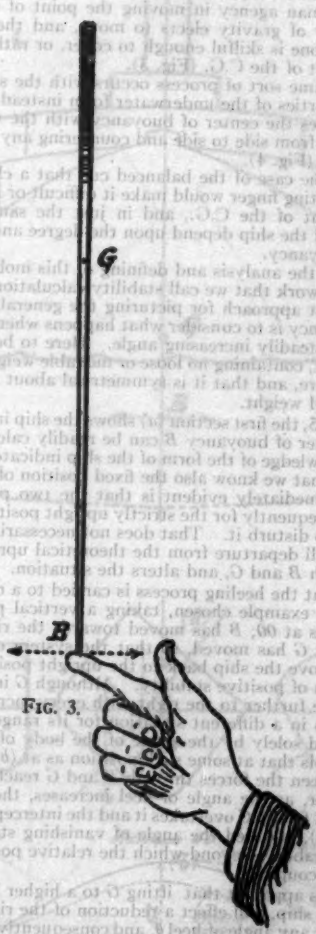


FIG. 3.

Although not in every particular a strict analogy, a simple illustration of the underlying principle of this somewhat peculiar characteristic of floating "stability" is afforded by the familiar process of balancing on one's hand some top heavy object such as, say, a billiards cue. Here the tendency to fall over is countered by human agency in moving the point of support in whatever direction the center of gravity elects to move, and the upright attitude is retained so long as one is skilful enough to cover, or rather more than cover, the initial movement of the C.G. (Fig. 3).

Very much the same sort of process occurs with the ship, except that it is the geometric properties of the underwater form instead of any human intervention that provides the center of buoyancy with the very accommodating property of moving from side to side and countering any tendency of the ship to lean or heel over (Fig. 4).

It is evident in the case of the balanced cue that a close limitation in the travel of the supporting finger would make it difficult or impossible to counter any large movement of the C.G., and in just the same way the inherent stability qualities of the ship depend upon the degree and range of movement of its center of buoyancy.

In effect, it is to the analysis and defining of this mobility of the center of buoyancy that the work that we call stability calculations is mainly directed.

Probably the best approach for picturing the general effect of mobility of the center of buoyancy is to consider what happens when a ship is heeled over in still water to a steadily increasing angle. Here to begin with we suppose the ship to be intact, containing no loose or movable weights, so that its center of gravity is a fixture, and that it is symmetrical about the middle line plane both as to form and weight.

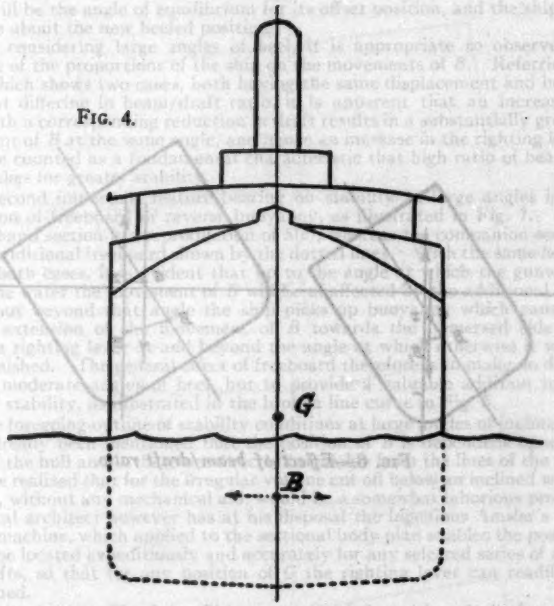
Referring to Fig. 5, the first section (*a*) shows the ship initially upright. The position of the center of buoyancy *B* can be readily calculated and fixed precisely from our knowledge of the form of the ship indicated on its "lines plan", and it is assumed that we know also the fixed position of the center of gravity *G*. All that is immediately evident is that the two points lie in the same vertical plane; consequently for the strictly upright position there is no system of forces tending to disturb it. That does not necessarily mean that the ship is stable, for a small departure from the theoretical upright obviously affects the position of both *B* and *G*, and alters the situation.

Suppose next that the heeling process is carried to a considerable angle, (*b*) shows that for the example chosen, taking a vertical plane of reference say through the keel as at *00*, *B* has moved towards the right or immersed side much further than *G* has moved, so that the system of forces constitutes a couple trying to move the ship back to the upright position. In other words there is a condition of positive stability. Although *G* in relation to the reference *00* must move further to the right with every increase of angle right up to 90 degrees, *B* is in a different situation for its range of movement to the right is conditioned solely by the shape of the body of the ship. The usual sequence of events is that at some such position as at (*b*), the righting lever or intercept *GZ* between the forces through *B* and *G* reaches a maximum value, and that thereafter, as the angle of heel increases, the movement *B* to the right slows up until finally *G* overtakes it and the intercept *GZ* vanishes. This, as illustrated in (*c*), is called the angle of vanishing stability and marks the angular range of stability beyond which the relative positions of *G* and *B* will create a capsizing couple.

In Fig. 5 (*b*) it is apparent that ifting *G* to a higher level *G*, say by adding top-hamper to the ship, will effect a reduction of the righting lever *GZ* by an amount $GG \sin \theta$ at any angle of heel θ, and consequently reduce both the righting lever and the range of stability as illustrated by the dotted curve in the diagram. Conversely of course, lowering *G* will increase the righting lever and range.

It is here assumed that the ship is symmetrical as to weights with *G* on the middle line plane, but Fig. 5(*b*) incidentally may be taken as illustrating the effect of any eccentricity of *G*. If *G* be moved off center on to a line through

FIG. 4.



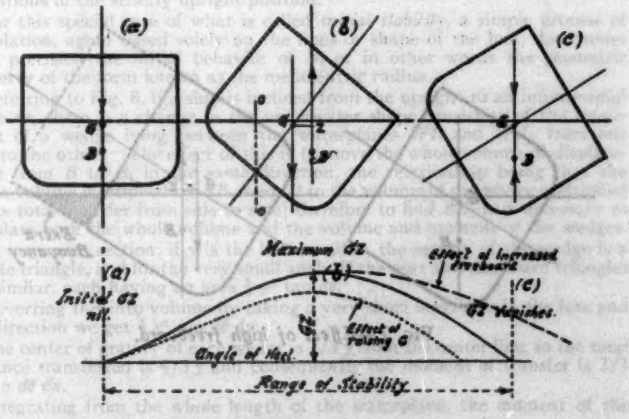


FIG. 5.—*Stability at large angles.*

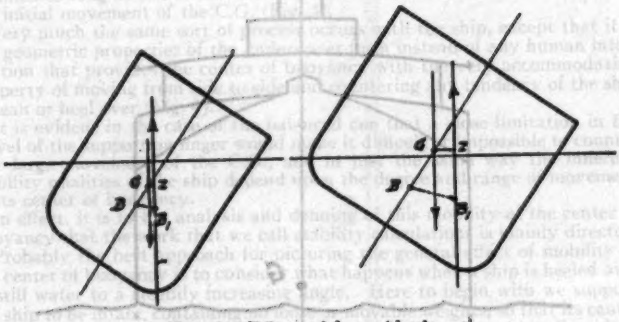


FIG. 6.—*Effect of beam/draft ratio.*

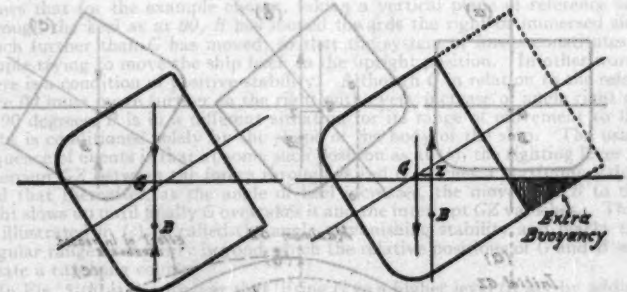


FIG. 7.—*Effect of high freeboard.*

BZ then, subject to a certain reservation as to the height of G, the attitude shown will be the angle of equilibrium for its offset position, and the ship will be stable about the new heeled position.

While considering large angles of heel, it is appropriate to observe the influence of the proportions of the ship on the movements of B. Referring to Fig. 6 which shows two cases, both having the same displacement and height of G, but differing in beam/draft ratio, it is apparent that an increase of beam with a corresponding reduction in draft results in a substantially greater movement of B at the same angle, and hence an increase in the righting lever. It can be counted as a fundamental characteristic that high ratio of beam to draft makes for greater stability.

The second important feature bearing on stability at large angles is the proportion of freeboard or reverse buoyancy, as illustrated in Fig. 7. Here the left-hand section is a reproduction of 5(c), whereas the companion section has the additional freeboard shown by the dotted lines. With the same height of G in both cases, it is evident that up to the angle at which the gunwhale enters the water the movement of B will be unaffected by the additional freeboard, but beyond that angle the ship picks up buoyancy which causes a marked extension of the movement of B towards the immersed side and creates a righting lever at and beyond the angle at which otherwise it would have vanished. The general effect of freeboard therefore is to make no difference at moderate angles of heel, but to provide a valuable addition to the range of stability, as illustrated in the broken line curve in Fig. 5.

In the foregoing outline of stability conditions at large angles of inclination, it has already been mentioned that the position of B is dependent upon the shape of the hull and is therefore directly calculable from the lines of the ship. It will be realized that for the irregular volume cut off below an inclined waterline this, without any mechanical aid, would be a somewhat laborious process. The naval architect however has at his disposal the ingenious Amsler's integrating machine, which applied to the sectional body plan enables the position of B to be located expeditiously and accurately for any selected series of angle and drafts, so that for any position of G the righting lever can readily be ascertained.

Referring back to Fig. 5, it will be appreciated that although the foregoing process can be used to calculate the righting lever with reasonable accuracy at quite small angles of inclination, it cannot define satisfactorily the stability conditions in the strictly upright position.

For this special case of what is called *initial stability*, a simple process of calculation, again based solely on the lines or shape of the hull, determines with precision the initial behavior of B, or in other words the geometric property of the form known as the metacentric radius.

Referring to Fig. 8, if a ship is inclined from the upright to an infinitesimal angle $d\theta$, there is a change in the underwater shape consisting of the movement of a wedge lying between the waterplanes WL and W_1L_1 from one side to the other. The effect of this is to move the whole volume of displacement from B to B_1 in the same direction, the relationship being that the whole volume multiplied by BB_1 is equal to the volume of the wedge multiplied by its total transfer from side to side; therefore to find BB_1 it is necessary to calculate both the whole volume and the volume and moment of the wedges.

At any cross section, if y is the half-breadth, the section of the wedge is a simple triangle, and for the very small angle $d\theta$ the port and starboard triangles are similar, each having an area $\frac{1}{2}\, y^2 \tan d\theta$.

Coverting this into volume by taking a very short length dx in the fore and aft direction we get $\frac{1}{2}\, y^2 \tan d\theta\, dx$.

The center of gravity of each wedge is $2/3\, y$ from the center line, so the total distance transferred is $4/3\, y$ and consequently the moment of transfer is $2/3\, y^3 \tan d\theta\, dx$.

Integrating from the whole length of the waterplane, the moment of the complete wedges is $2/3 \tan d\theta \int_0^l y^3\, dx$.

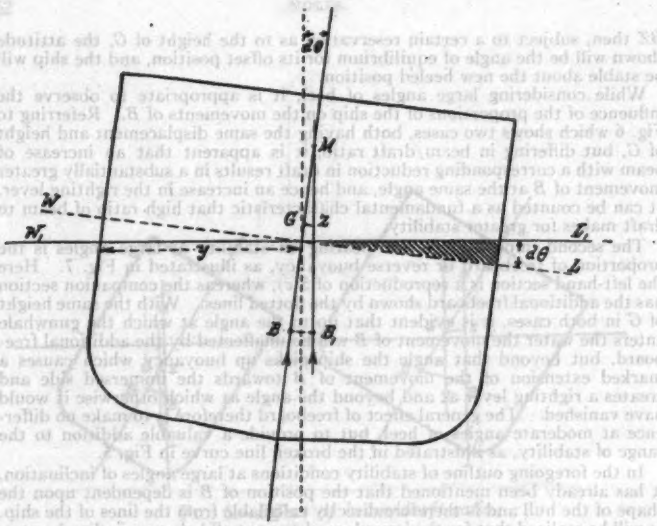


FIG. 8.—*Initial stability transverse metacentre.*

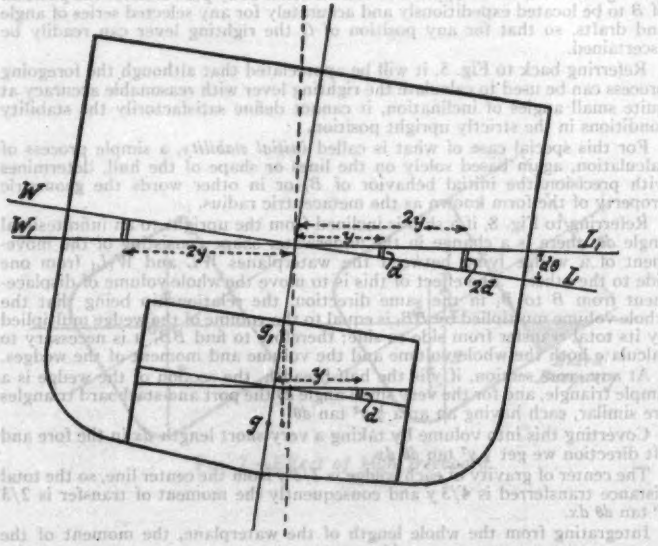


FIG. 9.—*Moment of inertia. Effect of free surfaces in tanks.*

Now $2/3 \int_0^l y^3 dx$ is the moment of inertia (I) of the waterplane about its central axis, and this can be calculated by Simpson's or similar rules, so the moment of the complete wedges is $I \tan d\theta$.

If ∇ is the total volume of displacement $I \tan d\theta$ is also equal to $\nabla \times BB_1$ or $BB_1 = \frac{I}{\nabla} \tan d\theta$.

For a very small angle BB_1 is at right angles to the center line plane, and the metacentric radius $BM = \frac{BB_1}{\tan d\theta}$. Therefore $BM = \frac{I}{\nabla}$ and as the position of B can be readily calculated from the lines of the ship, the position of M is located, and is known as the transverse "metacenter".

For small changes of angle for which the port and starboard wedges are substantially identical, M is practically a fixed point through which the vertical force of buoyancy acts, in other words the height of M defines the initial movement or behavior of B. The system of forces behaves as though the center of buoyancy were at M instead of at B or B_1.

Hence the relationship between G and M is the criterion of initial stability, and so long as G is located below M or, as we say, the ship has a positive GM, a resisting couple will be created by any departure from the upright position. The distance between G and M is called the "metacentric height".

The stability of a completely submerged body such as a submarine is a special case. Inasmuch as it has no waterplane, the center of buoyancy has none of the mobility which occurs in the surface craft due to changes in immersed form by transference of wedges. Hence a righting couple can only exist when the center of gravity is located below the actual center of buoyancy.

In the preceding calculation it will be seen that the height of the metacenter M above the centroid B of the immersed volume is obtained by dividing the "moment of inertia" of the waterplane by the total immersed volume.

It is well to have a clear picture of what is conveyed by the expression "moment of inertia", for it reappears in connection with the important question of free surfaces of liquids in ships' tanks and their bearing on stability. Translated into everyday language the moment of inertia ($2/3 \int_0^l y^3 dx$ in the mathematical code) conveys a measure of the plane's resistance to being pushed around.

Looking at the waterplane wedges reproduced again in Fig. 9, if we consider any small element of the plane at a distance y from the central axis, the process of forming the wedge generates a small volume d which is lifted from water to air on the one side while its counterpart is pushed down from air into the water on the other side, hence both resist this movement by the moment yd. Now another similar element at distance $2y$ will form double the volume and its moment of resistance will be quadrupled, that is to say, proportional to y^2. By the same reasoning the resistance of any other element will be proportional to the square of its distance from the center line. The addition of these moments of all the elements comprising the complete waterplane area is the moment of inertia, or in other words the moment represented by the transfer of the complete wedge volume from one side to the other.

This leads to consideration of the detrimental effect of free surfaces of liquids in partially filled tanks. Referring again to Fig. 9, in which such a tank is illustrated, it will be seen that a small angle of inclination causes a transfer of wedges on the water surface in the tank, which have a family resemblance to the wedges at the ship's waterplane, the wedge angles being exactly the same. The vital difference of course is that whereas the surface moment yd of an element of the ship's waterplane on the immersed side is an upward buoyancy moment, the corresponding moment of a similar element on the tank surface is a downward weight moment and one cancels the other. The effect therefore is as though an area equivalent to the tank area were cut out of the ship's waterplane, and its moment of inertia and the resulting height of M correspondingly diminished.

Alternatively, after adding the tank liquid as though it were a solid weight at g, the loss of metacentric height due to the free surface may be assessed as an increase in the height of G instead of a lowering of M; the liquid in the tank behaving as though its center were located at a higher position g_1, following similar reasoning as for the ship's metacenter.

It is immediately apparent that the transverse dimension of the free surfaces must be kept as low as possible.

If for instance the tank in Fig. 9 were divided by a center line bulkhead the free surface loss would be only one quarter that of the undivided tank, or if divided into three, the loss would be only one-ninth; so it may be regarded as a fundamental that all tanks should be so divided and operated as to keep the free surfaces to a minimum.

Knowledge of the initial stability properties of a ship's form as presented in Fig. 8, that is to say the calculated heights of the metacenter and displacements ar various drafts, provides a means of obtaining the actual metacentric height (GM) of a completed ship by a practical operation known as the *inclining experiment.*

The information obtained from the experiment is of particular value, for until the ship is completed the height of its center of gravity is a matter of more or less accurate estimation. But when by the inclining experiment the actual metacentric height is determined, it follows from our knowledge of the height of M that the height of G is also definitely located to check against the estimated data.

Briefly the routine of the experiment is as follows:—The ship is loosely moored to float freely in still water, all weights on board are secured against movement, and tanks preferably either empty or pressed full so that there are no disturbing free surfaces. A weight of inclining ballast sufficient to give the ship a small but measurable angle of heel is put on board, initially disposed centrally and so positioned that it can be conveniently traversed across the deck through a measured distance. The drafts and density of the water are carefully measured so that the displacement (Δ) is known precisely.

The ballast weight or weights are then moved across the deck to and fro as many times as desired and the resulting angle of heel measured as accurately as possible by means of a long plumb-line or other device.

Referring to Fig. 10, at the commencement of the experiment the ship is upright, its total weight (Δ) is known and also the height of the metacenter (M). When the ballast is moved across the ship and the angle of heel (θ) recorded, it is known that the center of gravity of the whole system is now off center and must lie somewhere in the new vertical plane through M. But the distance that the center of gravity has moved (GG_1) is equal to $\frac{w \times x}{\Delta}$ and as G has moved parallel to the direction x, GG_1M is a right-angled triangle in which $GG_1 = GM \tan \theta$.

Hence GM is readily obtained from the equation $\frac{w \times x}{\tan \theta \times \Delta}$. When G has been ascertained in the inclining condition of the ship, it is a matter of simple calculation to assess the metacentric height in any desired service condition.

Fig. 10 (a) shows the customary disposition of inclining ballast on the deck; the sequence of operations being to move, say, the port groups in turn over to starboard, return them to the original position and follow by moving the starboard groups to port, pausing between each move to observe the angle of heel.

Fig. 10 (b) shows an alternative method using, when space permits, a larger single group disposed initially on the center line, and making five shifts in the sequence shown. The latter arrangement is rather more expeditious if a crane is used to move the weights as it avoids transferring the slings.

For recording the angle of heel the traditional method is to use two long pendulums of 15 ft. or upwards, slung in hatchways or other sheltered positions. These are quite accurate in ideally still conditions, but if the ship is at all "restless", are prone to give irregular oscillations which militate against accurate measurement of the angle.

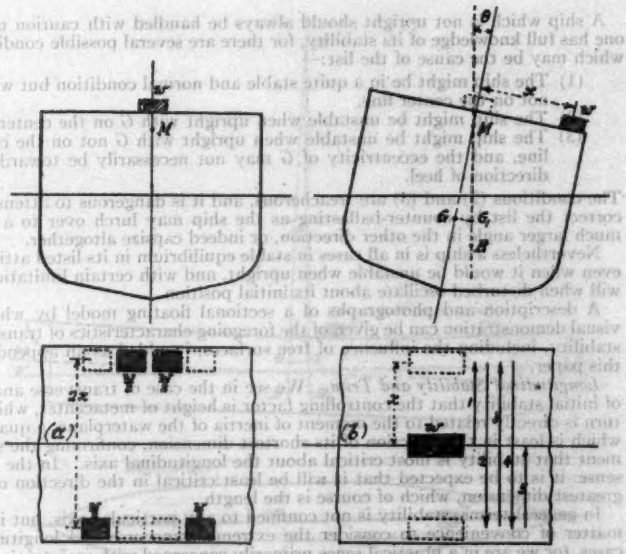


FIG. 10.—*Inclining experiment.*

Various types of angle recorders have been devised in substitution for the long pendulum, and Fig. 11 shows an example designed by the writer some twenty years ago, and which since that time has been in constant use. This tanometer, which takes only two or three minutes to set up and when dismantled is easily handled and transported, is a short-period heavy pendulum giving direct and accurate reading of the tangent of the angle of heel, amplified to a scale equivalent to a pendulum 16 ft. in length. Experience has shown that it can be used, particularly in small ships, in conditions where the ordinary long pendulum would be ineffective. Another practical advantage is that the tanometer can usually be more conveniently positioned so that the movement of weights and observations of angle are all under the supervision of one operator.

It is important to remember that the elementary metacentric rule defining the relation between the angle of heel and the eccentricity of $G - \tan\theta = \frac{GG_1}{GM}$ —which is used in the inclining experiment analysis, assumes that M is a fixed joint and that the sides of the ship in the vicinity of the waterplane are arcs of circles. This rule is substantially correct only when the ship has appreciable positive GM, but in the case of conventional ship forms which are more or less wallsided, the relationship does not hold good when the GM is very small, zero, or negative. In such cases a more complex mathematical analysis is entailed, and from all points of view it is preferable that a ship should be ballasted, if necessary, to be in a reasonably stable condition when inclined.

One or two points in connection with transverse stability are worthy of brief comment.

The upright attitude of a ship of symmetrical form is direct evidence that G is on the center line, and that G cannot be at a higher level than M.

A ship which is not upright should always be handled with caution unless one has full knowledge of its stability, for there are several possible conditions which may be the cause of the list:—

(1) The ship might be in a quite stable and normal condition but with G not on the center line.
(2) The ship might be unstable when upright with G on the center line.
(3) The ship might be unstable when upright with G not on the center line, and the eccentricity of G may not necessarily be towards the direction of heel.

The conditions (2) and (3) are treacherous, and it is dangerous to attempt to correct the list by counter-ballasting as the ship may lurch over to a very much larger angle in the other direction, or indeed capsize altogether.

Nevertheless a ship is in all cases in stable equilibrium in its listed attitude, even when it would be unstable when upright, and with certain limitations it will when disturbed oscillate about its initial position.

A description and photographs of a sectional floating model by which a visual demonstration can be given of the foregoing characteristics of transverse stability, including the influence of free surfaces, is added as an appendix to this paper.

Longitudinal Stability and Trim. We see in the case of transverse analysis of initial stability that the controlling factor is height of metacenter, which in turn is directly related to the moment of inertia of the waterplane; a quantity which is least in the direction of its shortest dimension, confirming the statement that stability is most critical about the longitudinal axis. In the same sense, it is to be expected that it will be least critical in the direction of the greatest dimension, which of course is the length.

In general terms, stability is not confined to any particular axis, but it is a matter of convenience to consider the extreme transverse and longitudinal cases, for we are in a practical sense primarily concerned with maintaining the ship in an upright attitude transversely, and an even balance or trim longitudinally. Since the equation $BM = I/V$ holds true for any oblique axis, the intermediate stability conditions may be resolved into transverse and longitudinal components.

Fundamentally the principles of longitudinal and transverse stability are alike, the practical distinction being that in the longitudinal direction we are seldom concerned with anything more than relatively small changes of attitude. A large change of trim of, say, 14 ft. in a 400 ft. ship, for instance, represents only an angular change of 2 degrees, so in all but exceptional circumstances the analogy to transverse analysis need not be pursued beyond the conditions of initial stability.

Referring to Fig. 12, if it be assumed that a waterline WL represents the static normal flotation attitude in still water, and that the ship experiences a small angular change to waterline W_1L_1, without alteration to displacement, there are repeated all the features of waterplane wedges, mobility of B, metacentric point M, etc., similar to those in the transverse analysis. Owing, however, to the very much greater lengthwise dimension, the numerical value of the moment of inertia of the waterplane, which is now referred to a transverse axis through its centroid, is so large that M is at a very high level. From the former reasoning, this is an index of a wide range of mobility of B in the longitudinal sense, consequently a very large restoring couple is generated by quite small angular change. It is evident that adequacy of longitudinal stability is so little in question that, for normal ship proportions, it can be considered to be an inherent property, and the problem is almost wholly a concern of the relation between attitude, or what we call "trim", and the longitudinal position of G.

In common usage the term "stability", unless specially qualified, is intended to mean transverse stability.

Trim is ordinarily measured by reference to the draft marks at the ends of a ship. For example a "trim by stern" of two feet means that the after draft exceeds the forward draft by two feet. If by a change in the position of

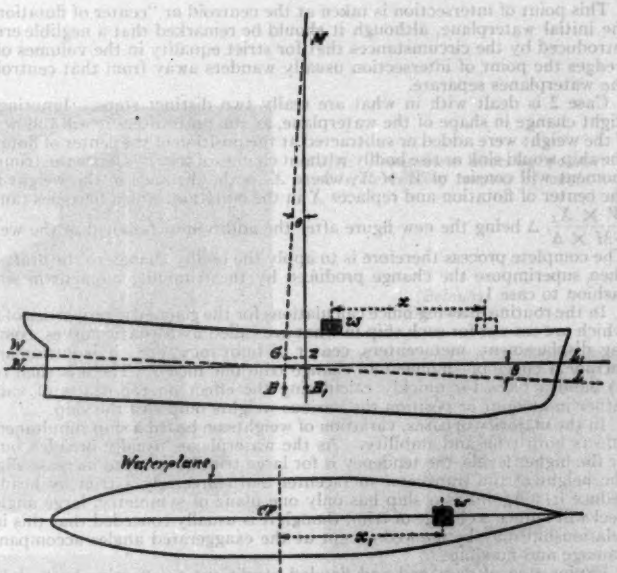


FIG. 12.—*Longitudinal metacentre and trim.*

weights on board the figures are reversed, the ship would be said to experience a four feet "change of trim". When the forward and after drafts are the same figure, the ship is said to be "on even keel".

Now it is a simple matter to avoid eccentricity of G in the transverse direction and keep a ship upright, but there are bound to be in service all sorts of changes in fuel, cargo disposal, etc., which shift G longitudinally, and one practical application of a knowledge of the longitudinal metacentric height is to enable the corresponding change of trim to be assessed.

There are two frequently recurring cases:—

(1) Shifting a weight already on board.

(2) Adding a weight to, or removing one from the ship.

Case (1) is immediately recognizable by its resemblance to the inclining experiment operation already discussed, and as it is legitimate with small angles to regard the longitudinal M as a fixed point, a precisely similar formulation is applied viz:—

$$GM = \frac{W \times X}{\tan \theta \times \Delta} \text{ or } \tan \theta = \frac{W \times X}{GM \times \Delta}.$$

GM and Δ being known, this gives θ as the angular change of trim. In translating the angular change of trim into alteration of drafts on the marks, the alteration will be plus at one end and minus at the other, and as can be seen in Fig. 12 the proportion depends upon the point of intersection of the two waterplanes WL and W_1L_1

This point of intersection is taken at the centroid or "center of flotation" of the initial waterplane, although it should be remarked that a neglible error is introduced by the circumstances that for strict equality in the volumes of the wedges the point of intersection usually wanders away from that centroid as the waterplanes separate.

Case 2 is dealt with in what are really two distinct steps. Ignoring the slight change in shape of the waterplane, as the draft alters it will follow that if the weight were added or subtracted at the position of the center of flotation, the ship would sink or rise bodily without change of trim. Hence the trimming moment will consist of $W \times X_1$ where X_1 is the distance of the weight from the center of flotation and replaces X in the equation, which becomes $\tan \theta = \frac{W \times X_1}{GM \times \Delta}$, Δ being the new figure after the addition or removal of the weight. The complete process therefore is to apply the bodily change to the drafts and then superimpose the change produced by the trimming moment in similar fashion to case 1.

In the routine drawing office calculations for the geometric properties of form which are set out for each ship in what are called hydrostatic curves, comprising displacement, metacenters, center of buoyancy, etc., it is customary to include a curve of "moment to change trim one inch". There is then ready to hand a basis for quickly calculating the effect on the drafts of varying either in amount or position the various weights on board the ship.

In the majority of cases, variation of weights on board a ship simultaneously affects both trim and stability. As the waterplanes usually broaden out aft at the higher levels, the tendency is for large trim by stern to increase slightly the height of the transverse metacenter and conversely a trim by head will reduce it. Again, as a ship has only one plane of symmetry, large angles of heel will induce a change of trim, though it is usually conceded that this interrelationship may be ignored except at the exaggerated angles accompanying damage and flooding.

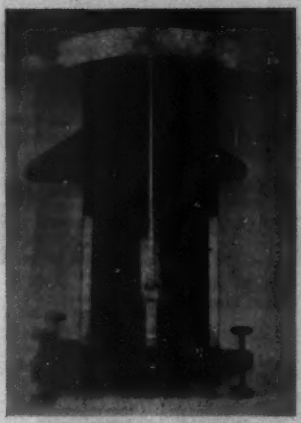
Exploration of damaged and flooded conditions as affecting both stability and trim is a matter of primary importance in modern naval architecture, and the preparation of very comprehensive calculations is compulsory in the design stage of a passenger ship, in order to implement the national and international standards of safety set out in the Merchant Shipping Convention Act.

In the realm of merchant shipping one naturally hopes that in all this we are catering for nothing more than a remote contingency. On the other hand, in naval ships, combat damage and consequent flooding must be dealt with as part of the order of existence, and in this field it will be appreciated that both design investigations into a large variety of damaged conditions and subsequent operational control, entail a tremendous volume of sometimes laborious calculations.

Demonstration Flooding Models. Bearing on this branch of the subject, the members may be interested in an example of the extension of scale model technique to the combined problem of stability, trim, and damage control. Just prior to the war, the writer chanced to be associated with shipbuilding in miniature in the production for the British Admiralty of large-scale floating models of capital ships and fleet aircraft carriers.

As will be seen from the photographs (Figs. 13 and 14), these models admitted of simulating and studying a wide variety of possible damaged conditions both with respect to the effect of flooding on the attitude of the ship, and its correction, thus demonstrating visually what could otherwise have been portrayed only by a bewildering array of figures. Hitherto under secrecy ban, the Admiralty now has kindly consented to the publication here of a brief description of these models.

Constructed of sheet brass, the models were very carefully made true to scale, both as to shape, weight and position of center of gravity; and were subdivided into a veritable honeycomb reproducing all the watertight compartmentation of the actual ship. By a system of simply operated flood valves and vents, any individual compartment or range of compartments



(Left): Front view showing pointer and scale reading tangent of angle.

(Below): Dismantled.

(Left): Back view showing pendulum and linkage.

FIG. 11.—*Tanometer for measuring small angles of heel.*

FIG. 13.

FIG. 14.

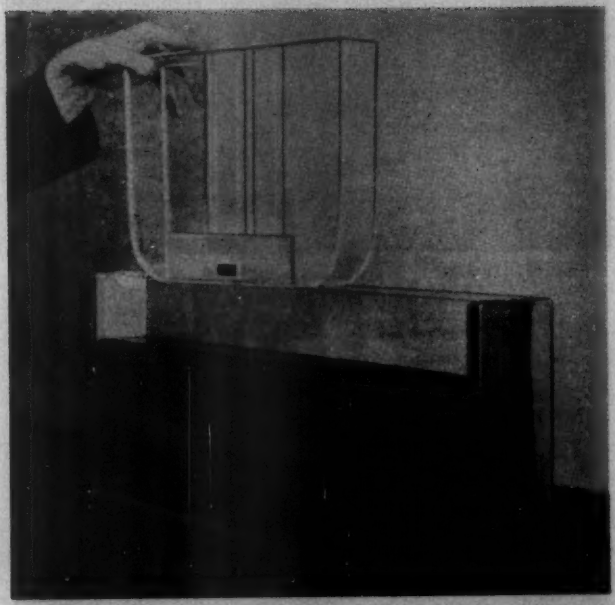


FIG. 17.—THE MODEL AND TANK.

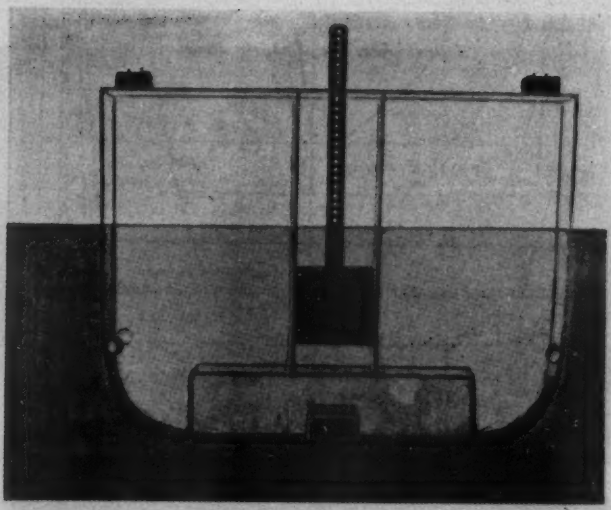


FIG. 18.—WEIGHTS SYMMETRICAL—LOW G. GOOD STABILITY.

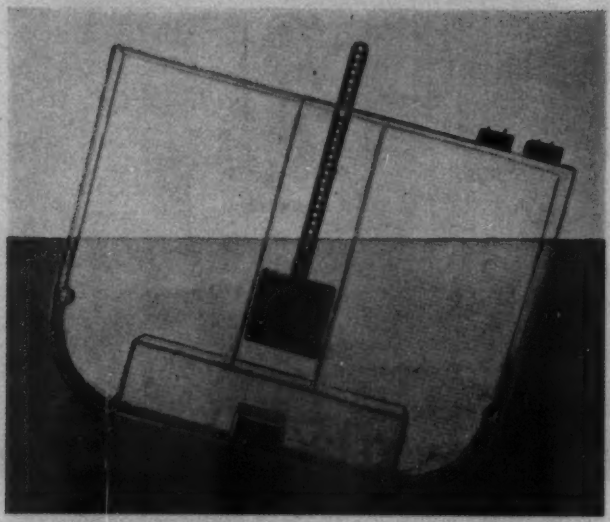


FIG. 19.—WEIGHTS ECCENTRIC—LOW G. HEEL 15°.

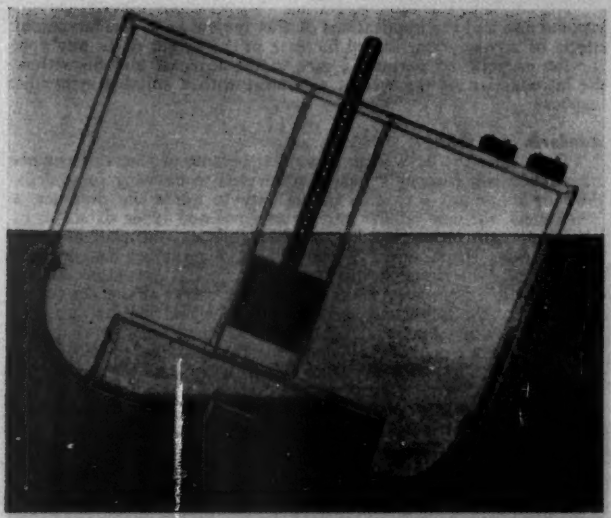


FIG. 20.—AS FIG. 19, BUT WITH "FREE SURFACE" IN BALLAST TANK. HEEL 20°.

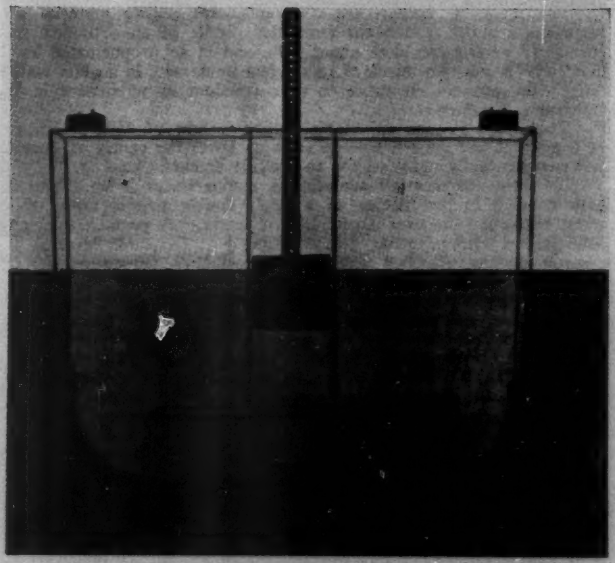


FIG. 21.—WEIGHTS SYMMETRICAL—HIGH G.

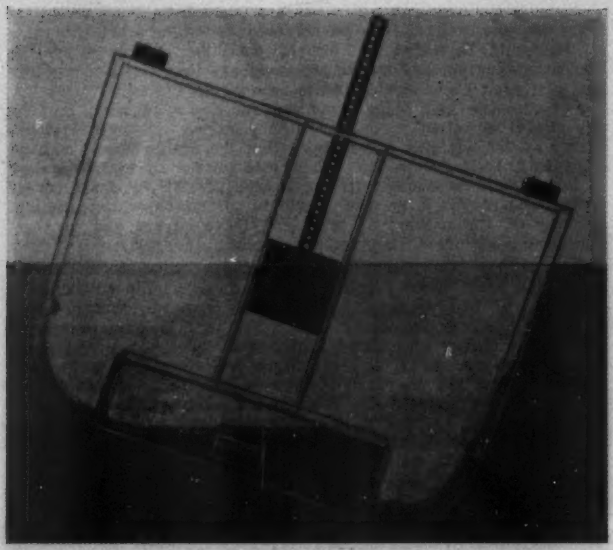


FIG. 22.—AS FIG. 21, BUT WITH "FREE SURFACE" IN TANK. UNSTABLE IN UPRIGHT POSITION. HEEL 18°.

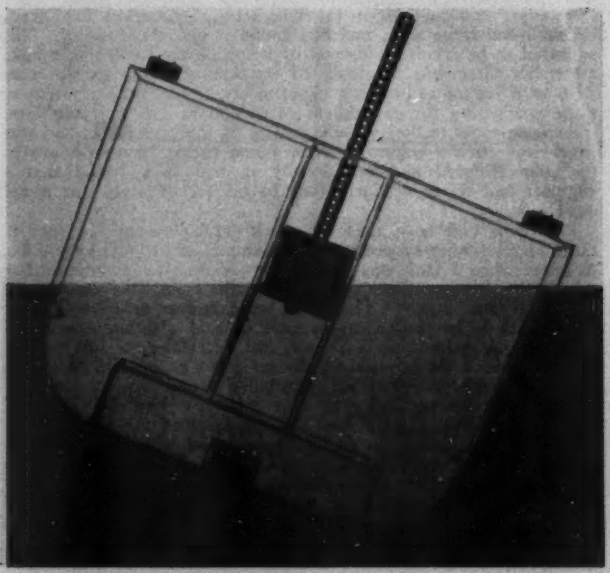


FIG. 23.—WEIGHTS SYMMETRICAL—VERY HIGH G. UNSTABLE IN UPRIGHT POSITION.

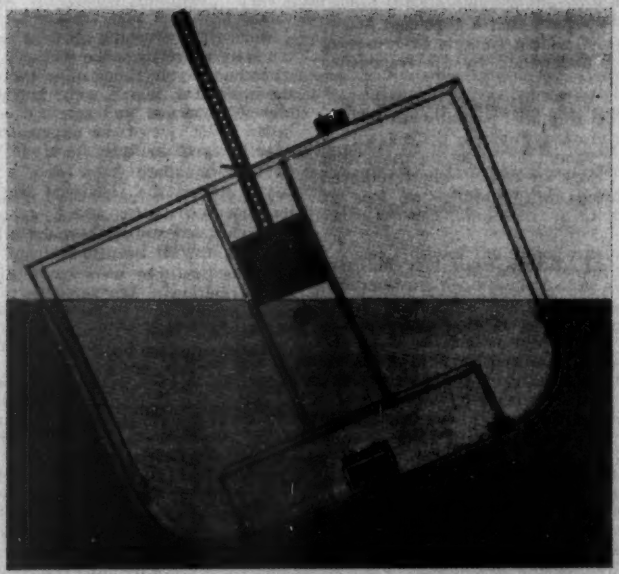
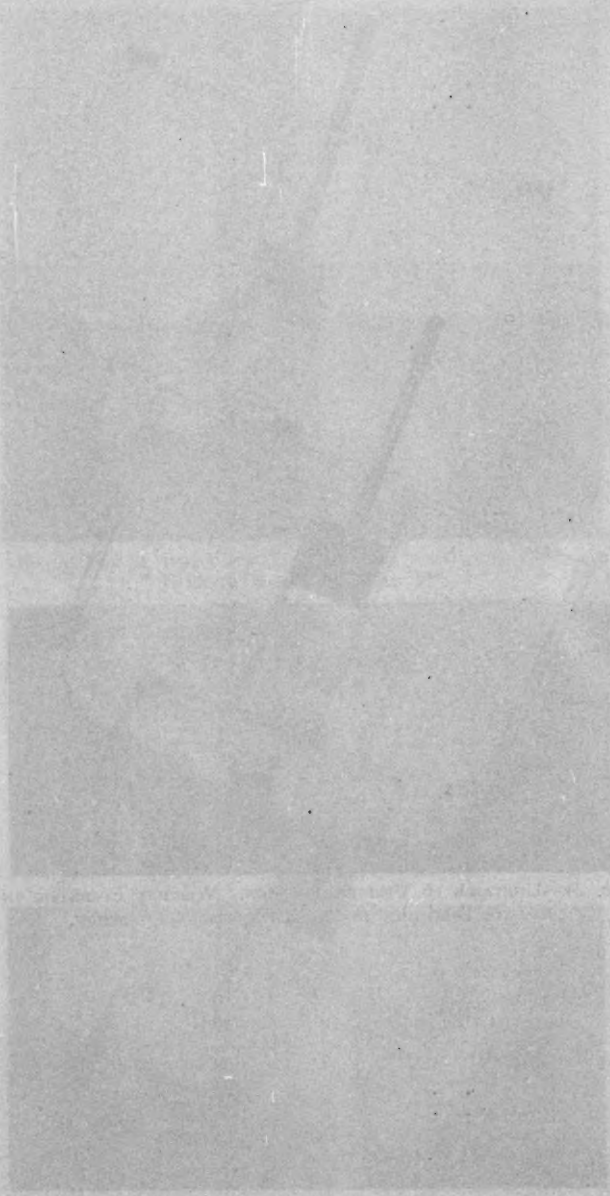


FIG. 24.—UNSTABLE IN UPRIGHT POSITION. WEIGHTS ECCENTRIC IN OPPOSITE DIRECTION TO LIST. DANGEROUS CONDITION.



could be flooded at will, and in order that ship conditions would be accurately portrayed each compartment contained a non-floodable canister proportioned to give the appropriate permeability. A proportion of the ballast was removable to give various initial conditions of loading. The model floated in a special glass-walled tank, and this was provided with an overhead gantry by which the model could be lifted from the water and turned over for draining. In passing it might be remarked that these models gave a very illuminating demonstration of the ability of our large naval ships safely to withstand a very extensive amount of both bodily and eccentric flooding.

Dynamical Stability. In the foregoing notes we have seen that a righting moment is generated by the separation of B and G in a lateral direction. A function of this moment is the work done in heeling the ship or what is called the "dynamical stability". This enters into calculations dealing with angle of heel under such impressed forces as the recoil of guns discharged broadside, and with the motion of ships among waves.

The curve of righting levers plotted as ordinates to a base of angle of inclination provides a simple method of assessing the dynamical stability. Converting the curve of righting levers into a curve of righting moments by multiplying by Δ throughout, we get a varying moment which has to be overcome in heeling the ship.

The work done in overcoming the moment up to any selected angle θ is $\int_0 \Delta \times GZ \times d\theta$, and as this expression is proportional to the area under the curve up to the angle θ, a derived curve of dynamical stability may be plotted by measuring the area of the statical stability curve by planimeter to a number of successive angles.

Relation between Stability and Rolling. The study of the rolling motion of ships among waves introduces a factor, the time element, which is absent in the geometric analysis of statical stability and the subject extends into a wide field of theory beyond the scope of a brief review. It is appropriate, however, to intrude a note or two on the general influence of stability qualities on the rolling characteristics of a ship.

In the first place, as a corollary to the forced heeling of a ship in still water as shown in Fig. 5 (*b*), it will be observed that those proportions and features which generate large righting moments and for which the direct index is a large metacentric height, are also those which increase the responsiveness of the ship to angular changes in the water surface itself.

This responsiveness is reflected in the time factor, for what is called the ship's natural period of oscillation, that is to say, its free rolling motion in still water is related *inter alia*, to the metacentric height; roughly speaking inversely as its square root. Hence large metacentric height contributes to "stiffness', characterized by quick, uncomfortable motion, while a small metacentric height results in a slow, easy motion much more to the liking of passengers and crew.

The waves also according to their size, pattern and angle of approach have a period of encounter, and it is largely the relationship between the respective periods of the ship and the waves that determines the behavior of the ship.

In the extreme case of a broad raft-like vessel with a very large metacentric height, the natural period will be very short and the ship's inclination will tend to keep normal to the varying slope of the wave profile, in other words the deck will remain parallel to the water.

This represents a condition of forced rolling where the ship moves with the wave period and not at all in its own natural period.

Turning to the other extreme of a deep and narrower ship form with small metacentric height, the rolling period among waves approaches more nearly to the ship's natural period and is consequently slower and more comfortable.

Long sustained and regular period in the waves might develop a forced roll more in accord with the time of the wave period, but such regularity in wave trains is seldom encountered.

The principal danger lies in the building up of a large amplitude of roll if there should happen to be close agreement or synchromism between the ship's

natural period and the period of encounter with the waves, but in general there is less probability of sustained synchronsim if the ship has a small metacentric height and consequently a long natural period.

All these considerations lead to the inevitable conclusion, confirmed also by experience, that small or moderate metacentric height is an essential objective in the pursuit of easy motion and sea-kindliness. Unfortunately, a conflicting issue arises in the part played by metacentric height when we come to consider damaged and flooded conditions and other special circumstances.

The angle of heel in, say, a cross-channel ship caused by deck passengers crowding to one side can only be minimized by providing substantial metacentric height which, unfortunately, would have the effect of making the ship a quick and uncomfortable roller.

Similarly the insistence by the authorities on strictly limited angle of heel in passenger steamers in assumed conditions of side damage where eccentric flooding would ensue, in order that lifeboat handling, freeboard and navigation shall remain effective, is tending towards an increase in metacentric height to the detriment of behavior in a seaway.

The middle course of compromise is no help, for in the choice between seasickness and safety there appears to be no alternative to regarding the latter as the *fundamental*.

Ship Stabilizers. No doubt it is in recognition of this that inventive effort in the field of naval architecture is still directed in search of some effective form of what are sometimes referred to as stabilizing devices, though a more appropriate name would be rolling dampers.

Numerous ideas have been produced and tried during the last three-quarters of a century with varying measure of success. The simple and familiar bilge keels have a marked damping effect on rolling, and are almost universally adopted other than for craft operating in sheltered waters. They have a double effect of reducing the amplitude of roll and increasing the mean period, though their effect is less pronounced when the ship has no headway than when it is travelling at speed.

Almost coincident with the early application of bilge keels there began the first development of the idea of anti-rolling tanks, which are in principle a deliberate introduction of free surface effects.

These anti-rolling tanks (see Fig. 15) consist of partially filled water chambers in the wings of a ship, permitting the flow of a body of water from side to side, which by virtue of being out of phase with the normal rolling period of the ship creates a moment tending to counter the ship's righting moment and damp the roll. Various applications of the anti-rolling tank principle based on the Frahm type enjoyed a considerable vogue, two notable cases being those in the Atlantic liners *Bremen* and *Europa*. However, the feature of reduction in righting moment and fear of the potential risk of increased rolling if the water and ship ran into phase have no doubt been responsible for the disappearance of anti-rolling tanks in recent times in favor of more positive roll-damping devices such as the gyroscopic stabilizer and the oscillating fins.

Gyro stabilizers of the Schlick and Sperry types are the culmination of a long process of development, the latter being exampled in the very large installation in the *Conte di Savoia* comprising three wheels weighing about 110 tons each, spinning at some 900 revolutions per minute. It is generally conceded that gyro stabilizers of adequate size may be expected to diminish the amplitude of rolling by something in the order of 50 per cent.

The oscillating fins of which the Denny-Brown stabilizer is a current example, is in quite a different category and produces a moment in opposition to the inclination of the ship by the force exerted on submerged protruding fins by the action of the water streaming past them much in the fashion of a rudder.

The fins are power-operated to change their attitude in sequence with the rolling of the ship, so that the force is always in the direction of damping the rolling. The action is governed by a small gyro-controller, and the fins are retractable, so that they may be drawn within the fair line of the hull when not in use. This stabilizer is very effective, but the damping force naturally

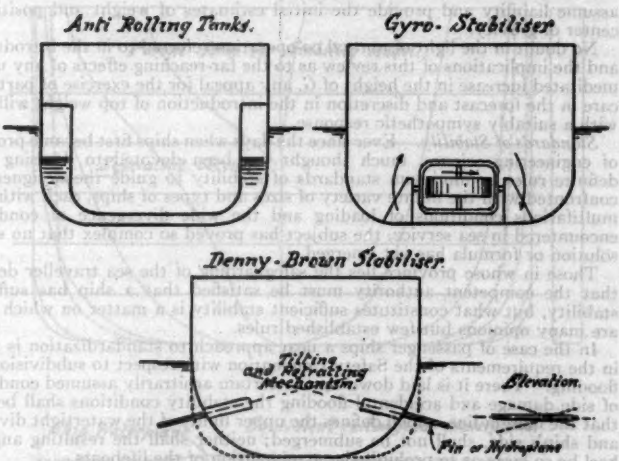


FIG. 15.—*Rolling dampers.*

diminishes with reduction in ahead speed, and is altogether absent when the ship is stationary, except for the additional bilge keel effect of the protruding fins. Positively activated stabilizers go a long way to improving the behavior and comfort of a ship among waves, but the irregularities of really violent wave motion and the extravagant forces and amplitudes generated by the condition of yaw-heel still leaves a tantalizing problem for the naval architect.

Center of Gravity. It will be apparent from any study of the mechanics of stability that the proper selection of proportions and forecasting of metacentric height is contingent upon an accurate knowledge of the position of the center of gravity. In presenting a brief account there is little choice but airily to dismiss it as having a "given" position, but unfortunately in the uncharitable realm of the general practioner the position is not given and has to be assessed by calculation.

In small and simple structures, the calculation of center of gravity may be a fairly straightforward and not particularly laborious process, but with increase in size and complexity of structure and contents of a ship, the location of G becomes less amenable to direct calculation and more a matter of estimation by comparison with existing types in which G has been definitely ascertained by an inclining experiment.

In the work-a-day initial processes of building say a large liner, the most disconcerting aspect of any dubiety as to the accuracy of the estimated position of G is that once the lines or form of the ship have been selected and the work of construction set in motion the naval architect has, as it were, "burnt his boats", for the position of M and other geometric properties are then irrevocably fixed. At that early stage the conception of much of the contents is inevitably general rather than detailed and in large measure undecided as to precise material or weight, so contemplation of subsequent accretion of weight particularly in top-hamper associated with the contents of tiers of passenger accommodation and superstructure can make the harmless symbol G into what someone feelingly described as a naval architect's nightmare!

There is here a divided responsibility inasmuch that, of the total weight, a substantial proportion is contributed by the propelling machinery, electrical and other installations for which the naval architect's engineering colleagues

assume liability and provide the initial estimates of weight and position of center of gravity.

No doubt in the light of mutual co-operation referred to in the introduction and the implications of this review as to the far-reaching effects of any unpremeditated increase in the height of *G*, any appeal for the exercise of particular care in the forecast and discretion in the introduction of top weight will meet with a suitably sympathetic response.

Standards of Stability. Ever since the days when ships first became products of engineering science, much thought has been devoted to devising some definite rules or minimum standards of stability to guide the designer, but confronted with the infinte variety of sizes and types of ships, each with their multifarious conditions of loading and the wide divergence of conditions encountered in sea service, the subject has proved so complex that no simple solution or formula has yet emerged.

Those in whose province lies the safeguarding of the sea traveller demand that the competent authority must be satisfied that a ship has sufficient stability, but what constitutes sufficient stability is a matter on which there are many opinions but few established rules.

In the case of passenger ships a near approach to standardization is found in the requirements of the Safety Convention with respect to subdivision and flooding. There it is laid down that in certain arbitrarily assumed conditions of side damage and accidental flooding the stability conditions shall be such that the margin line, which defines the upper limits of the watertight divisions and ship's side, shall not be submerged; neither shall the resulting angle of heel be so large as to prohibit the manipulation of the lifeboats.

In general the problem resolves into reconciling two conflicting considerations. It has been already emphasized that small metacentric height is desirable from the point of view of easiness at sea, but on the other hand the stability must cater for:—

(1) Accidental flooding and resulting heel.

(2) Preventing undue heel by crowding of passengers to one side.

(3) Preventing undue heel when encountering strong side winds or in turning at speed under helm.

The solution lies in providing the least metacentric height that will satisfy these three conditions.

Just twenty years ago a special committee appointed by the American Marine Standards Committee issued a report on their very comprehensive survey on the subject of stability and loading of ships, and recommended a procedure and formulae for determining a minimum standard of stability. One recommendation which found universal acceptance was that full technical information on the stability qualities of every ship should be put at the disposal of the operating personnel, who hitherto has been left more or less to their own devices, and this is a course which is now a legal obligation in the terms of the Merchant Shipping Acts.

Appendix.

Floating Sectional Model for illustrating Transverse Stability. The model is constructed of transparent plastic material and resembles a short section cut from the middle of an orthodox ship form. It weighs complete about 7½ lb. and is so proportioned as to float, with about two-thirds of its volume immersed, in a glass walled tank with sufficient clearance to give freedom of movement in the transverse direction.

There is a central adjustable ballast weight of 4¼ lb. which can be pegged at any desired level to raise or lower the center of gravity, and two smaller weights on top to provide eccentric loading.

To illustrate free surface effect a compartment is built into the bottom much in the fashion of a ship's double bottom tank, which when half-filled contains about ¾ lb. of water. A portable block of solid ballast also weighing

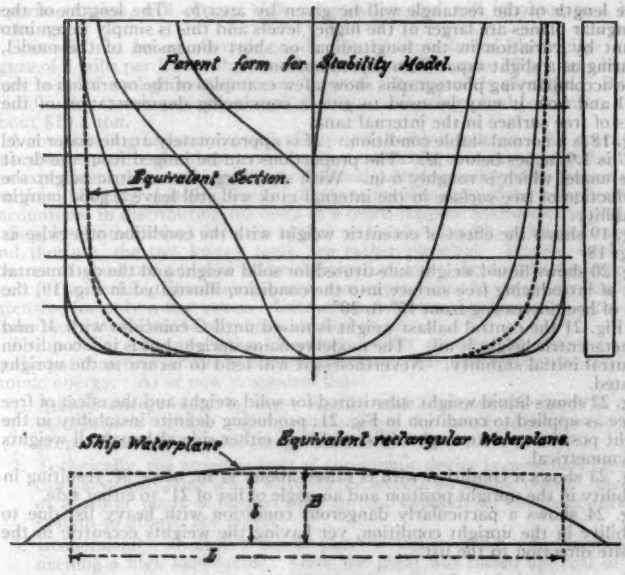


FIG. 16.

¾ lb. is situated at the same level as the centroid of the half-filled tank and can readily be substituted for the water in the tank.

Although for purposes of demonstration any arbitrary section would suffice, the model has actually been developed from a ship's body plan to constitute from the point of view of transverse stability, what might be termed an equivalent section which takes into account the shaped form. That is to say, at any draft the waterplane of the model has the same relative area and moment of inertia as the corresponding waterplane in the ship form, and consequently a similar height of metacenter (M) and mobility of (B) in the transverse direction. For static conditions therefore the model will closely reproduce to scale the angles of heel and behavior of the parent ship form of which the body plan is shown in Fig. 16.

In producing the equivalent section it is assumed that the usual waterplane areas and inertias have been calculated from the ship's lines and are already tabulated. Now if at the level of each curved waterplane an equivalent rectangular plane is substituted which has the same area and I, the volume ∇ and hence BM will be unaltered and subject to certain adjustment for length variation the single section bounded by the widths of these equivalent waterplanes will have substantially the same geometric properties as the whole curved form.

Referring to Fig. 16, the procedure is that knowing for the ship's waterplane I = area $\times B^2 \times$ a factor, the factor is readily calculable (X).

Also that for the equivalent rectangle I = Area $\times b^2/3$.

From which we must have $B^2 \times X = b^2/3$.

$$\therefore b = B\sqrt{3X}$$

The length of the rectangle will be given by area/b. The lengths of the rectangular planes are larger at the higher levels and this is simply taken into account by variation in the longitudinal or short dimension of the model, appearing as a slight taper from top to bottom.

The accompanying photographs show a few examples of the operation of the model and how it may be used to give a convincing demonstration of the effects of free surface in the internal tank.

Fig. 18 is a normal stable condition. M is approximately at the water level and G is 1.9 inches below M. The proportions can be judged from the draft of the model which is roughly 6 in. With such large metacentric height the introduction of free surface in the internal tank will still leave a good margin of stability.

Fig. 19 shows the effect of eccentric weight with the condition otherwise as in Fig. 18.

Fig. 20 shows liquid weight substituted for solid weight and the detrimental effect of introducing free surface into the condition illustrated in Fig. 19, the angle of heel increasing from 15° to 20°.

In Fig. 21 the central ballast weight is raised until G coincides with M and the metacentric height is nil. The model remains upright but is in a condition of neutral initial stability. Nevertheless it will tend to return to the upright if canted.

Fig. 22 shows liquid weight substituted for solid weight and the effect of free surface as applied to condition in Fig. 21; producing definite instability in the upright position and an angle of list of 18° to either side although all weights are symmetrical.

Fig. 23 shows a condition with G raised about ¼ in. *above* M, resulting in instability in the upright position and an angle of list of 21° to either side.

Fig. 24 shows a particularly dangerous condition with heavy list due to instability in the upright condition, yet having the weights eccentric in the opposite direction to the list.

INDUSTRY'S STAKE IN ATOMIC ENERGY.

This article by Robert Colborn is reprinted from Chemical Engineering for March, 1947. It gives a very good overall picture of the present organization for handling the development of atomic energy applications in this country.

Ever since Hiroshima, speculation has been rife on the implications of atomic energy on human society. Although many people are still inclined to act as though it were a private problem of statesmen, soldiers and physicists, alert engineers and businessmen are beginning to realize that atomic energy will have far reaching effects throughout industry. Research in atomic energy is fast widening and its results must be considered in any long range industrial planning. It should, therefore, be well to know the direction of this research and how it is being carried out.

Power Generation.

Commercial production of electric power from atomic engines is thought by many to be only about five years away. Experimental production of power on a substantial scale will be under way within a couple of years at at least three places—Oak Ridge, Schenectady, and Chicago. This is much sooner than was being predicted a few months ago and reflects promising results of preliminary investigations.

Power piles will be designed to operate at temperatures similar to those of a modern high-pressure steam plant. Fluids such as molten bismuth, sodium-potassium alloys or others which can reach high temperatures without creating high pressures, will first be heated in the pile and then used to generate high pressure steam. New piles of the Clinton laboratory at Oak Ridge and the Knolls laboratory at Schenectady are specifically designed for power production. The Argonne laboratory's new pile at Chicago, though intended for general

experimentation, will run at high temperature and will incorporate a heat exchanger.

Cost will not be an obstacle to the use of atom-fueled electricity. The figure of 8 mills per Kwh. which has been estimated is some 30 per cent higher than costs of coal-generated power in areas where coal is plentiful. But it would make atomic power economical in any area where coal costs more than about $10 a ton.

A complicating factor in costs is the question of byproducts. An operating pile always produces valuable radioactive materials or may perhaps be used for experimental, metallurigcal, or chemical purposes other than power production. Pricing of atomic power is likely to involve some of the complications encountered in distributing the costs of a multi-purpose hydroelectric project.

Supply of atomic fuel does not appear to be any obstacle either. Uranium and thorium, the two known fuels, are rather plentiful. Known world resources of good ore are said to be sufficient to meet the world's total power need for something like twenty years, and exploration for uranium has been intensive for only a few years. Extraction from low grade sources—granite, shales, seawater—has hardly been studied at all.

The peculiar characteristics of atomic power sources are perhaps more significant than cost factors in assessing the industrial effects of commercialized atomic energy. As of now it appears that:

1. Atomic energy may be most readily and economically used in large installations. Plant and operation costs for small plants are out of proportion to the useful energy produced.
2. Technically, the power output of any atomic device is extremely flexible, can vary easily from a few watts to the level of an atomic bomb. Output of a power plant is limited only by the capacity of the heat transfer system to absorb the power:
3. Economically, an atomic power plant would resemble a hydro plant in needing a high load factor. Once the plant was fueled up, cost of fuel replenishment would be a minor factor. Fixed capital charges, which continue no matter what the output, would be the large element in cost. However, labor cost is a larger factor than in conventional plants. Much of the labor would be in the "ash removal" chemical plant and could vary somewhat with the power output.
4. Most novel and perhaps most significant feature of an atomic power plant is that it is completely independent of transporation. Because its fuel input is measured in pounds rather than tons, it can be put wherever power is needed. Not only does this simplify problems of electricity transmission, but it can have a profound effect on the geography of all the heavy power-consuming industries. It might become economical, for instance, to make aluminum near the bauxite fields rather than transport bauxite to a power supply.

Isotopes.

Of wide significance is the manufacture of elements not found at all in nature—the so-called "radioactive isotopes." These are elements with the same chemical properties as the natural elements but possessing the additional property of emitting radiation, much as radium does. Radioactive forms of nearly every element can be manufactured in quantity in an atomic pile.

The research and medical value of radio-isotopes has been well publicized, but their potential usefulness in many industrial processes is not as widely realized. Two types of use are promising—as radiation sources and as tracers.

Production of radio-isotopes is still so small as to limit their use to laboratories, but they will be available to industry also within a year or two. Any wide use of power piles would make radio-isotopes an available industrial tool.

Developed in a country which, more than any other, stresses private enterprise as the key feature of its economy, this new and far-reaching technology was immediately and almost unanimously seen as a thing which had to be socialized. There was no serious dispute in Congress last year as to giving

the government complete control over the new industry; the May-Johnson and McMahon bills both did that. The fight was over the secondary question of what branch of the government should exercise the control.

More, the idea is being taken seriously that introduction of such a fundamental new factor into the world's economy is a matter for the organized efforts of the entire human race. The U. S. government is officially sponsoring a proposal to create an entirely novel form of political instrument for international socialization of the atom.

It's hardly surprising therefore, that confirmation of the membership of the Atomic Energy Commission set off a senatorial battle rather more bitter than that over the basic legislation. The confirmation question gave Congress and industry a second look at the problem after appreciation of the sweeping nature of the issues had become clearer.

The Socialized Industry.

By the terms of the McMahon Atomic Energy Act, every phase of atomic activity is under complete government control. Key activity is the manufacture of fissionable material. This is a complete government monopoly. It is illegal for a private person to own any fissionable material or any equipment capable of producing it. Only exception is that equipment capable of producing negligible quantities may be used for research purposes, subject to a government license. All patents in this field are wiped out. No patent can be granted on any discovery usable solely in production of fissionable material and any such patents previously existing have been cancelled. Moreover, all patents are nullified as far as production of fissionable materials is concerned; any patented article or process may be used without regard to the patent. Anyone who makes a discovery in this area is required to bring it to the commission.

Private ownership of uranium, thorium, or high-quality ores of these materails is still legal, but all dealings in them, once they leave the ground, are subject to a license from the commission. AEC may requisition or condemn such materials or any land containing deposits of them.

Private ownership and operation of devices using atomic energy is also legal —but only if the commission grants a license for them. It's entirely up to the commission whether it will grant a license; no standards are set except the public welfare and security and the maintenance of free competition.

The commission itself may also finance or carry on development of methods to utilize atomic energy. The law says nothing as to how the commission shall handle the applications it develops, except that any power produced may be used by the commission itself, turned over to other government agencies, or sold to public or private utilities under contracts providing for reasonable resale rates.

Either through ownership or through its licensing powers, the Atomic Energy Commission has complete control over the rate and the manner of introduction of nuclear technology into industry. And the law leaves the commission almost unlimited discretion. Congress did not intend this situation to be permanent. It's a temporary arrangement until the problems involved become clearer. For all its powers, therefore, there is a tentative aspect to every decision of the commission. Essentially it's an interim body, carrying on until the situation has settled down enough to permit long-range decisions. Inevitably, however, the present AEC will be setting patterns, creating vested interests, that can't help influencing the Congress when the time comes for another review of the situation.

The atom is going to be a political issue, and an important one, for a long time to come—taking its place alongside such economic-political questions as public power, taxes, labor relations. The international negotiations on the atom now making their sluggish way through the UN introduce another big element of uncertainty.

Pattern for Operations.

Eventually, the AEC will have to deal with a long string of economic and social questions—timing the commercialization of atomic processes, working out procedures, setting priorities, easing the impact on competitive industries, establishing prices. These are still several years away.

The commission is actively seeking answers to a lot of immediate questions. It needs a patent policy based on the novel and still uninterpreted provisions of the McMahon Act; a consulting group of patent experts is working on this now. The commission has a group of leading accountants trying to work out a decentralized financial system for it which will minimize red tape; objective is a large measure of autonomy for individual units within the limits of an annual budget.

On one major point, commission policy is fairly well established. AEC will continue the Manhattan District policy of operating all its plants and laboratories by contract, rather than as purely governmental agencies. When AEC took over, about 50,000 of the atom workers were on contractors' payrolls and only about 5000 were drawing government paychecks. This proportion is scheduled to decrease if anything.

In some cases the contractual relationship is pretty nominal. The contracting firm may have been especially organized for the job. Or the contractor's connection with the work may hardly extend farther than putting its name on a document and appointing a staff satisfactory to AEC. Here, the practical effect may be pretty close to that of a quasi-public corporation. In other cases, of course, the contractors are very much in the picture and are exercising a large voice in policy.

Terms of AEC contracts are still treated as secret. It is known, however, that they are on a non-profit basis—cost-plus-nothing or plus a fee of $1. Cost in the contracts, however, is interpreted rather more broadly than in the standard fee contract so as to protect the contractor against any possibility of out-of-pocket loss.

There's sone question how long the non-profit arrangement can be maintained. Many businessmen doubt that the commission can get the wide industrial participation it wants unless it provides for some return to its contractors.

The issue is sharpened by the precautions AEC is taking in an effort to prevent giving its contractors too much of an inside track on ultimate commercialization of atomic energy. Chief among these is the device of an advisory or consulting board for each contractor, representing a range of interested outside firms and assigned definite rights under the contract.

This idea was first developed by the university people who've been working out schemes for administration of research, but it's being inserted by AEC into its contracts with industrial firms. Thus the contract covering General Electric Co.'s operation of the Knolls laboratory not only specifies that members of the advisory committee must be given complete access to this work but also requires G. E. to appoint to its laboratory staff people nominated by the advisory firms up to a maximum of 15 per cent of personnel.

As a result, there's a feeling among some firms that under the present set-up they would be as well off represented on advisory committees as they'd be with all the headaches of actual management.

Some time in the 1950's the Atomic Energy Commission may turn out to be one of the most powerful economic planning agencies in the federal government. For the present, AEC is a long way from that. It is a research agency, expanding our knowledge of the underlying principles of nuclear energy and learning the technics of applying it.

The commission's research work now centers at four laboratories inherited from the Manhattan District. Three additional laboratories are under construction, and the commission has contracts for specific research projects with more than a dozen universities and research agencies.

The four functioning laboratories are: (1) Clinton Laboratory at Oak Ridge, operated by the Monsanto Chemical Co.; (2) Argonne National Laboratory at

Chicago, operated by the University of Chicago; (3) Berkeley Radiation Laboratory operated by the University of California; (4) Los Alamos Laboratory in New Mexico, also operated by the University of California. New laboratories in the works are the Brookhaven National Laboratory being built at Patchogue, Long Island, which will be operated by a group of eastern universities; the Knolls Laboratory at Schenectady, to be operated by General Electric; the Dayton Engineer Works near Miamisburg, Ohio, to be operated by Monsanto.

This assortment of facilities represents the remnants of the bomb project which AEC takes over. There are still a lot of decisions to be made as to how research will eventually be handled. But the way policy is shaping up suggests that the final pattern will include these elements:

PATTERN FOR RESEARCH.

1. The Los Alamos laboratory as a center for actual weapon development. Although this is a pretty definitely governmental job, it will probably continue to be operated, at least nominally, by the University of California, in accord with the AEC administrative policy of avoiding the straight civil service type of set-up.
2. Three or more laboratories operated by industrial firms and assigned pretty specific objectives. Thus the General Electric laboratory at the Knolls is to concentrate almost exclusively on generation of usable power from atomic piles. Monsanto's Clinton laboratory will probably emphasize the chemical problems of pile operation and the production of isotopes. Work to be done at the Dayton works has not been revealed. Associated with each of these industrial labs there will be an advisory or consulting group of other industrial firms, who will thus be enabled to keep in touch with what is being done and to make what contribution they can.
3. Most of the more fundamental research will center around a group of "regional" or "national" labs. Much of the outside contract research will eventually be handled through them, and they will be intended to guide and finance the bulk of university nuclear research. Argonne and Brookhaven are the two such units approved so far. Another will very likely be formed on the West Coast around the present Berkeley lab. Later there may be a southern lab and one in the northwest.

Central theme at the Argonne lab in Chicago is the design of atomic piles—at a somewhat more fundamental level than the engineering approach involved in the power production program. Immediately engrossing problem is to work out the features and characteristics of piles which run hot, at temperatures in the neighborhood of the 900 deg. level of modern steam plants.

But including and stretching beyond this job are a whole range of problems to be investigated. Relative merits of different moderators such as graphite, heavy water, beryllium. Characteristics of piles using uranium with various degrees of enrichment in U-235 or plutonium. The metallurgy of uranium itself, so little studied before the war that not even the melting point of metallic uranium was known with any precision. Mechanical, chemical and nuclear characteristics of structural pile materials under the conditions of an atomic fire will be studied.

Early indications are that the Brookhaven National laboratory will focus on basic nuclear research. It will work on such things as the nature of the different sub-atomic particles; the properties of the mysterious "binding force" that holds them together in the atomic nucleus and whose release is the source of the energies of the atom.

If and when a national laboratory is set up on the West Coast it will probably inherit from the Berkeley Radiation Laboratory its interest in the design and construction of particle accelerators ("atom smashers"). These rank with the atomic pile as principal tools of nuclear research.

Although each of the national laboratories seems destined to evolve or be assigned its own field of concentration, each of them also serves a second

function—as a regional center for the nuclear research carried on in all the neighboring universities and research institutions.

In the first place each laboratory will be a pool of equipment for the universities. Negotiations are now in progress between AEC and the military services to insure that in future most major government-owned equipment will be centralized at the labs, where it will be available to a broad group.

In general government financing of university atom studies will channel through the regional labs. AEC will not simply farm out to the universities work it wants done; it will also help finance almost any worthwhile research in the field.

The Regional Role.

At each of the labs there will be at least three different levels of work under way: First will be the research carried on by the permanent laboratory staff. This is where the field of concentration of the particular laboratory will have most influence. Second will be so-called "sponsored" programs of research. These will normally originate when someone on a university staff brings in a line of work he'd like to undertake—and the work seems to fit rather definitely into the AEC program.

Still a third will be when one of the universities might bring in a job of research which involves the principles or the equipment of nuclear physics but has no very clear connection with AEC work. Then, if the project looks sound from the scholarly point of view, the lab might make it a "participating" project.

This whole system, still in an early and somewhat tentative stage, has been drawn in terms of university problems. Little thought has yet been given to the possibility of use of the regional laboratories by industrial researchers.

Although the national laboratories have both a national and a regional role, it's their regional function that has dictated their administrative arrangements. Within very broad limits, the labs will be run, their policies and programs will be determined, by the neighboring universities.

Legalistically, this can take several forms. Brookhaven is to be run by a specially-formed corporation—Associated Universities, Inc.—organized by nine eastern universities. Because the midwestern state universities lack legal authority to organize corporations, the Argonne laboratory is being run by a single institution, the University of Chicago, subject to a sort of board of directors from 24 other institutions.

This approach was not originated by AEC. It's the result of several years of discussion and thought by university people connected with the bomb project. It's their answer to the problem of reconciling the obviously necessary government financing of nuclear research with the freedom from regimentation which they also want.

Industry Participation.

The atom is one bandwagon that U. S. industry has shown little eagerness to get aboard. Of the hundreds of companies involved in the bomb project during the war many are unattracted by the post-war atom program; some have pulled out. Notable example is Du Pont which gave up the operation of the Hanford plutonium works last fall; similarly Eastman, which operated the now largely abandoned electromagnetic separation system at Oak Ridge, shows little interest in taking on other atomic work.

Among the few companies which, postwar, are broadening their atomic activity are Fairchild Aviation, General Electric, and Monsanto. Fairchild is handling AAF research on atomic propulsion of aircraft (the so-called NEPA project). General Electric is operating Hanford and will run a new AEC laboratory at the Knolls outside Schenectady. Monsanto took over new management of the Clinton laboratory at Oak Ridge in mid-1945 and is building a smaller laboratory for the AEC at its Dayton research center.

One reason for disinterest may be the inevitable socialization of atomic technology—inevitable not only because of its military and potential economic

significance but because of the huge sums which still have to be spent, without much hope of an early return, in order to exploit it.

Industrial firms thinking about an aggressive program of atomic work, are necessarily gambling. They may eventually find themselves forced out of the field. They may never get to a position where they can make some money from it.

One reason for the strong position which General Electric is assuming in the atomic field is that, for G. E., the gamble is safer than for most firms. Officials of this company feel that the investment of executive effort is justified if only by the probability that exploitation of atomic energy will increase the world availability of electric power. With its strong position in the electric industry, the company's convinced that anything which increases the use of electricity can't help increasing the market for its products.

GENERAL ELECTRIC CO.

G. E. can see more direct commercial business in the atomic offing. The ultimate decision as to who shall build and operate atomic piles may throw them to utility firms, to a government agency, or to an international body. But, regardless, the odds are that there'll be profitable business in furnishing the specialized and complicated associated equipment—the heat exchanges, the control devices, the chemical equipment. G. E. is determined to accumulate early the know-how that will make it the logical supplier of that equipment.

When G. E. assumed management at Hanford, the transition was smooth even though major changes are scheduled. One change is the establishment of an engineering and design unit. Du Pont never had one at Hanford, made use of its strong central engineering division in the East. G. E., which doesn't centralize its engineering to the same extent, wants a unit at the plant. A small fundamental research unit has also been established at Hanford, chiefly to serve as liaison with the research people at Schenectady.

Another new project is a school of nuclear engineering at Hanford. One man has been picked to head it, full time, and he'll draw on the plant technicians for faculty. The school will train both Hanford people and outsiders. All these changes reflect one basic change in operating policy—the carrying on of a development program at Hanford.

As it takes over on a peacetime basis, therefore, G. E. faces a long range program of modifying the whole operation. The ideal would be a set of piles from which most of the by-product power could be recovered, through which uranium could be circulated in an efficient and automatic manner, and with which would be associated a more or less automatic chemical plant which would do an economical and reasonably complete job of getting out the plutonium while extracting or recirculating the other components.

Just outside Schenectady, construction is now getting started on the $20,000,000 Knolls atomic power laboratory. The first of the 200 or so scientists and about 600 technicians who will eventually work there are being recruited. The operation should be rolling by mid-1948.

Owned by AEC and operated by G. E., the layout will include at least one atomic pile as well as physics, chemistry, metallurgy and engineering facilities.

Although the Knolls atomic power lab will do much of the research work in connection with Hanford, its primary job is to develop techniques for utilization of atomic power—both in equipment of the central station type and for ship propulsion. Typical problem is that of the heat transfer fluid. A fluid must be chosen which has good nuclear properties. This means it must not absorb neutrons and so interfere with the chain reaction. It must not be transmuted by the energies of the pile into a less desirable substance.

Mechanically and chemically the fluid must be such that it can be handled, inside the pile, in pipes of a material which itself will have good nuclear properties and will retain its strength and chemical characteristics in the face of the great heat and fierce radiations of the pile. Since the fluid may not be water, its thermodynamic properties will need a lot of study for heat exchanger design. Its corrosive action, if any, on other materials will have to be investigated. Extremely reliable means of pumping and valving it under conditions of great radioactivity will have to be found.

Monsanto's Part.

Monsanto Chemical Co., AEC's other big post-war contractor, is taking a much longer gamble on the atom than G. E. Where General Electric can look forward to specific profitable business out of almost any likely pattern of use for atomic energy, Monsanto has to look a lot harder to see much assurance of eventual profit out of its operation of the Clinton and Dayton laboratories for the Atomic Energy Commission.

The company operates its atomic projects under the same no-fee, no-patent type of contract, but its feeling is that this can't be permanent, that AEC will have to turn to a more liberal sort of arrangement if it's going to get the wide participation of industry that it needs.

Nature of Monsanto's atomic work at Dayton has been described only as fundamental research continuing a wartime project. Known features of the Clinton laboratory at Oak Ridge include the most powerful experimental pile in the country and a large "hot lab" for chemical work with dangerously radioactive materials. Clinton developed many of the chemical processing methods used at Hanford for extraction of plutonium and is now the center for manufacture of radioactive isotopes.

Newest project at Clinton is the construction of a high temperature pile for power generation—the first such project to be announced. There's never been any public definition of the division of effort between Clinton and the Knolls on power generation. As a matter of fact there's plenty of room for parallel activity without duplication. The researchers are anxious to try out dozens of different types of pile.

But perhaps it can be assumed that Monsanto's background will be especially valuable in dealing with the chemical problems of uranium and plutonium circulation through the pile and extraction of fission products.

At any rate, it's along those lines that Monsanto foresees an eventually profitable role for itself in the atom business. It's convinced that there will have to be room eventually for proprietary rights in nuclear discoveries. And though Monsanto may never operate commercial piles, there's always the possibility of a commerce in ideas similar to the substantial business the company now does in licensing chemical processes.

Again, there's the whole little-explored field of application of radiation to chemistry, which it should pay a chemical concern to be in on. In the pharmaceutical field, in which Monsanto is closely connected, there'll be work to do in synthesizing complex compounds containing radioactive elements for medical tracer or therapeutic use.

RECENT DEVELOPMENTS IN ALTERNATING-CURRENT TURBO-ELECTRIC SHIP PROPULSION.

This article by G. O. Watson of Lloyd's is reprinted from the Journal and Proceeding of the Institution of Mechanical Engineers for March, 1947, and gives an interesting comparison of recent A. C. electric drive design

Introduction. Propulsion by turbo-electric machinery has made impressive strides during the past decade and particularly during the war period; it is therefore opportune to review some recent achievements. In mere numbers the efforts in the United States are outstanding and make impressive reading, the figures for turbo-electric ships for the 6½-year period, January 1939 to June 1945, being:—

Type	Number	Aggregate gross, tons	Aggregate shaft, horsepower
Transport, P2	6	99,284	123,000
Transport, S4-SE2-BD1	32	192,000	211,200
Transport cargo, S4-SE2-BE1	28	169,652	184,800
Tanker, T2-SE-A1	438	4,514,721	3,171,120
Tanker, T2-SE-A2	40	423,148	400,000
Tanker (Private interests)	7	75,165	42,210
Totals	551	5,473,970	4,132,330

These figures refer to merchant vessels actually completed and do not include those still under construction or naval vessels.

While the figures for the United Kingdom are relatively insignificant, account must be taken of the entirely different wartime programs of the two countries. The figures are as follows and, the contracts having been placed in the concluding stages of the war, these ships are mostly still under construction:—

Type	Number	Aggregate shaft, horsepower
Bel ships	4	30,000
Tankers	3	39,000
Passenger	1	13,000
Cargo liners	4	36,000
Totals	12	118,000

Early Developments. Before discussing more recent ships, it would perhaps be well to review early history in order to get a correct perspective. Credit for much of the early pioneer work must be given to the late Mr. Henry A. Mavor, whose perseverance and enthusiasm attracted much attention. About 1911, largely at his own expense, he fitted out an experimental launch, the *Electric Arc*, which, although equipped with Diesel engines and not entirely successful, nevertheless demonstrated the possibilities of electric drive with alternating-current machines. His work eventually attracted the attention of the United States Navy Department, who commissioned his services, with the result that in 1913 the U. S. A. collier *Jupiter*, the first ship fitted with turbo-electric alternating-current drive, went into service. She had twin screws, of 5500 Shp. and was subsequently renamed *Langley*. Converted into an aircraft carrier, she remained in service until sunk by enemy action in the Pacific during the recent war, after a wonderful life—for a pioneer—of over 30 years.

Shortly afterwards (in 1914), the S. S. *Mjolner*, using Ljungström turbines, was completed in Sweden followed in subsequent years with some fourteen or fifteen similar vessels.

The United States pursued this line of development and up to the end of 1944 had completed more than 900 turbo-electric plants, totalling nearly eight million shaft horsepower. Outstanding among these were the aircraft carriers *Saratoga* and *Lexington*, each with four screws, totalling 180,000 Shp. In merchant shipping outstanding examples were the liners *President Hoover* and *President Coolidge*, each 26,500 Shp. on two screws.

This recital of achievements would be incomplete without reference to Mr. W. L. R. Emmet. He contributed very largely to the pioneer work in this sphere, although his amibtion to apply the mercury turbine did not materialize.

On the British side, the field was entered in 1918 with the S. S. *Wulsty Castle* followed by the more noteworthy S. S. *San Benito* in 1921. The latter deserves special mention because, together with the S. S. *Cuba* completed in the United States about the same time, it represents the first use of the synchronous motor for propeller drives, and all subsequent equipments have been based on this type. Compared with the asynchronous or induction type of motor, unity power factor and higher efficiencies are obtainable.

British developments date from 1918 and, including those under construction, now total 263,000 Shp., which does not include exhaust turbine schemes, in which steam from reciprocating engines is passed through an exhaust turbine supplying power to a motor on the propeller shaft.

Mention has already been made of the Ljungström turbine applications in Norway and Sweden and to complete the picture we must include one built in Japan, four in France, four in Germany, and two in Holland.

Prior to 1920, all alternating-current installations employed the asynchronous or induction motor. Adjustment of speed was obtained by a combination of variable resistance in the rotor circuit and by varying the speed

and thereby the frequency of the turbo-alternator, though in some installations squirrel-cage motors were used and speed control was entirely by varying frequency. In the larger ships, such as battleships, use was also made of change-pole motors. Time will not permit entering into details of the many interesting features of these earlier ships, but sufficient has been said to indicate the type of equipment employed.

Subsequent to 1920, the synchronous motor has been employed universally.

Synchronous Motor Drives. The synchronous motor, as is well known, has negligible starting torque and the method used for starting and reversing depends on the incorporation of a squirrel-cage winding in the pole faces of the rotor so that the motor starts and reverses as an asynchronous or squirrel-cage induction motor.

Methods of Control. To obtain the best torque conditions for maneuvering, it is necessary to reduce the frequency of the system and the ratio chosen is usually about one-quarter of full speed, the controls being interlocked to ensure the proper sequence of operation. With normal excitation on the alternator, the stator voltage would collapse under the heavy starting current (four to five times full-load current) and the excitation is therefore boosted to approximately twice the normal when maneuvering. The sequence of operation from stop is as follows:—

(1) Close the direction switches, i.e. "ahead" or "astern". This connects the stators of the alternator and motor for correct direction of rotation.

(2) Energize the alternator field with double normal excitation.

(3) When the instruments indicate that the motor and alternator are at approximately synchronous speeds, excite the motor and reduce the alternator excitation to normal. The propeller will now be running at approximately one-quarter speed and further speed adjustment is made as required.

These operations are carried out by means of three levers, interlocked to ensure correct sequence of operation or alternatively by a single handwheel on which all the operations are carried out in the correct sequence. To reverse direction the three levers or the handwheel are brought to stop, the direction switches changed and the sequence repeated. The only variation in this method is that in some recent equipments the alternator field is not switched off in the stop position and the direction switches must therefore be suitable for operation under load conditions.

Excitation. One of the drawbacks of this system is the necessity for super-excitation of the alternator during maneuvering as it requires auxiliary D.C. power being available at all times to supply the extra current. In a typical 13,000 Shp. equipment, the normal excitation is of the order of 27 Kw. (330 amp. at 82 volts). During maneuvering this becomes 107 Kw. (660 amp. at 163 volts), and when the field windings are cold may reach 750 amp. or 122 Kw., but it is, of course, only a short-time demand—say, 3 or 4 seconds—on a simple maneuver up to about two minutes on a full-speed reversal.

The chief variation between one equipment and another is in the method of obtaining the extra excitation and methods are by no means standardized. Systems which have been used comprise the following:—

(1) Three-wire D.C. system. This was practically standard at one time in United States schemes. The three-wire system is used for ship's auxiliaries, 220 volts across the outer wires for power purposes, and 110 volts to the mid-wire for lighting, etc. For super-excitation the field was connected to the outers and for mornal running it was transferred to the mid-wire.

(2) Variable voltage exciter.

(3) Booster. If the ship system is 110 volts it is boosted to 220 volts when maneuvering but if the system is 220 volts a negative booster is employed to reduce the voltage for normal running.

A variation of method (3) comprised the use of a three-wire balancer set with direct-coupled boosters making a four-unit set. It was used in the twin screw equipment of the S. S. *Strathnaver* and *Strathaird* in which the auxiliary D.C. supply was 220 volts. The balancer provided 110 volts for motor excitation and the boosters gave the required excitation for the alternators.

Having now reviewed very briefly the main principles of operation on the electrical side we can consider some recent equipments in detail.

Recent Installations in U. S. A. Outstanding, not only in the turbo-electric field but also in relation to marine propulsion in general, is the equipment of United States tankers of which the S.S. *J. W. Van Dyke* is the prototype. Built for the Atlantic Refining Company in 1938 to the requirements of their consulting engineer, Dr. Lester M. Goldsmith, their performance represents a distinct advance in marine propulsion and several sister ships are now in operation.

Particulars of the hull are as follows:—

Displacement	23,898 tons
Length (between perpendiculars)	520 feet
Breadth, moulded	70 feet
Draught	29 feet, 6 inches
Block coefficient	0.780
Designed speed	13.25 knots

As a result of tank tests at Hamburg and at the Washington Model Basin it was decided that approximately 5000 Hp. would be required and steam at 600 lb. per sq. in. and 825 deg. F. was decided upon as affording the greatest economy. The third ship of this series, the S. S. *E. J. Henry*, has a steam temperature of 910 deg. F. An outstanding consideration was that of rapid discharge of cargo and electrically driven pumps of approximately 1000 Hp., having a total rated capacity of 10,500 gal. per min., are provided.

Special precautions were taken in the layout to prevent impure water entering the boilers, drained water subject to the possibility of contamination being re-evaporated. The turbines are arranged for three bleed stages. Three turbine-driven boiler feed pumps are provided, though only one is required in normal operation, and the exhaust from this is returned to the second bleed stage. Boilers are of the straight-tube header type with air preheater but without economizers, and this type was selected with a view to low maintenance charges on account of simplicity of design.

The results obtained from the log of voyage No. 12 of the S. S. *J. W. Van Dyke* are as follows:—

Shaft Hp.	5,080
Total steam per shaft Hp.-hr.	8.92 lb.
Percentage of total heat from fuel to shaft	22.7 per cent.
Total fuel per shaft Hp.-hr.	0.591 lb.
Percentage of fuel for auxiliaries	13.3 per cent.
Fuel per shaft Hp.-hr. for propulsion only	0.512 lb.

The auxiliary D.C. generating sets are electrically driven from the main propulsion generator from 60 down to approximately 52 cycles per second, when the speed governor of the auxiliary steam turbine opens up and the auxiliary turbine takes over. The equipment is designed to incorporate modern power station practice, and complete automatic combustion control is provided, including superheat control. This is so effective that, as proved by chart recordings, the superheat temperature actually drops when load is reduced.

The turbo-alternator is rated 4600 Kw., 60 cycles, 2300 volts and 3600 Rpm. The turbine is of the impulse type, fifteen stages with bleeding at the third, seventh, and twelfth, and has a speed range of from 900 to 3780 Rpm. with corresponding propeller speeds of 22.5 to 94.5 Rpm.

The electric controls have been simplified by combining reversing and field control in one lever so that for operating the line contactors and field switching only one lever is used, and this is interlocked with the speed levers. Contrary to usual practice, an extra lever is provided for emergency use, which works directly on a throttle valve, and this permits hand control up to 75 per cent of full speed.

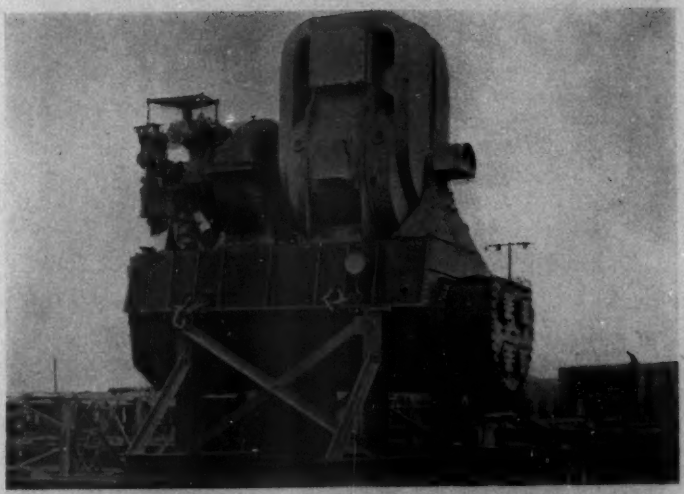


FIG. 2.—ASSEMBLED PROPULSION 6600 HP. TURBINE GENERATOR SET AND CONDENSER COMPLETE AS USED IN THE U.S.A. TURBO-ELECTRIC TROOPSHIPS.

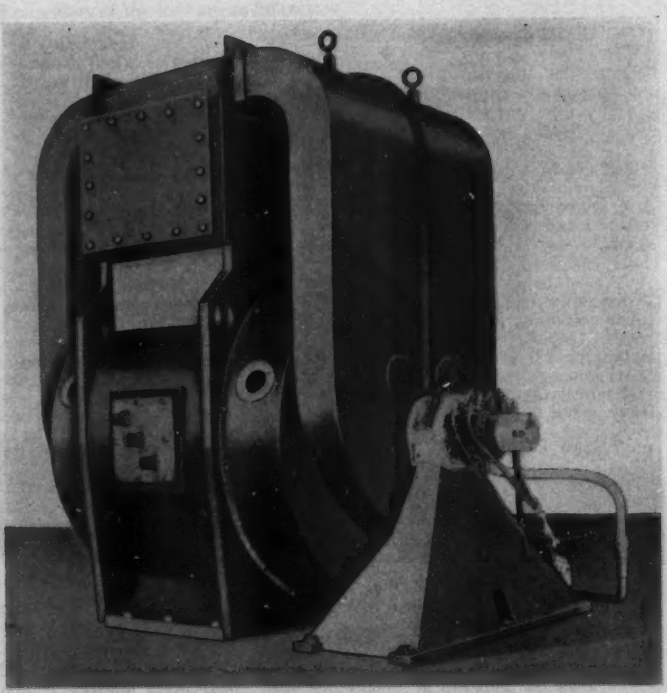


FIG. 3.—CLOSE-UP VIEW OF THE GENERATOR ALONE SHOWING THE COOLER AND CONNECTING DUCTS MOUNTED ON THE MACHINE SHOWN IN FIG. 2.

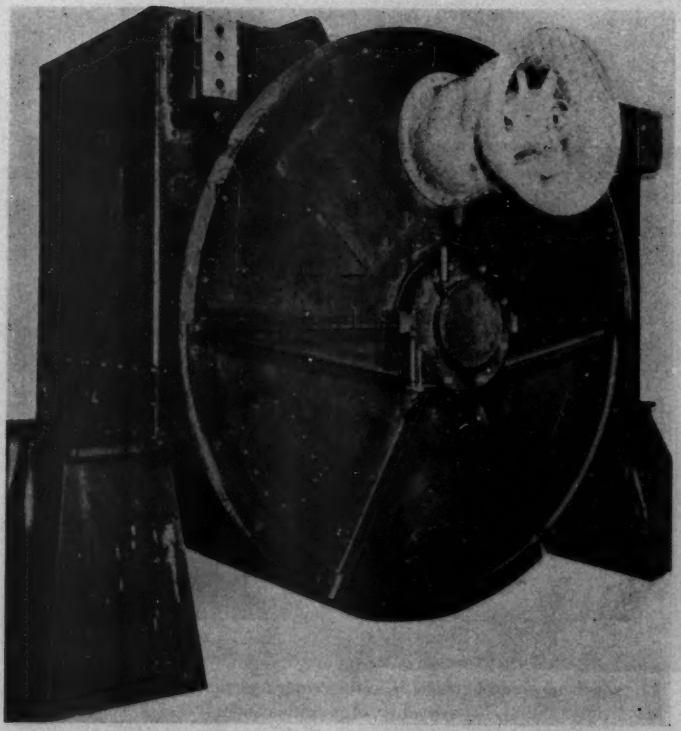


FIG. 4.—U.S.A. COAST GUARD CUTTER 4000 S.H.P. PROPULSION MOTOR AND COOLING SYSTEM.

The two air coolers are mounted on top of the motor frame, one at each side. Air taken from the motor room passes through the machine and then returns directly through the air coolers into the motor room.

FIG. 5.—ENGINE ROOM CONTROL BOARD ON 4000 S.H.P. U.S.A. COAST GUARD CUTTER.
The propulsion control cubicle is in the center and the auxiliary switchboards are adjacent on either side.

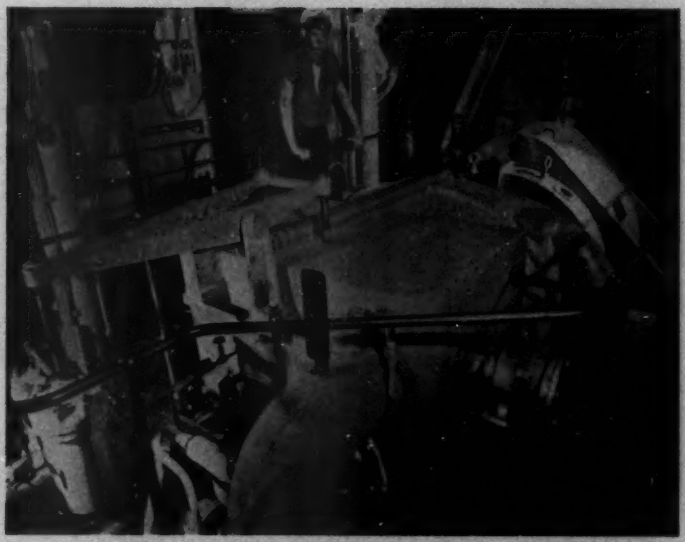


FIG. 6.—PROPULSION TURBO-GENERATOR ON 4000 S.H.P. U.S.A. COAST GUARD CUTTER.
In this case the generator air cooler is mounted in the structure below the generator. The control board is in the background.

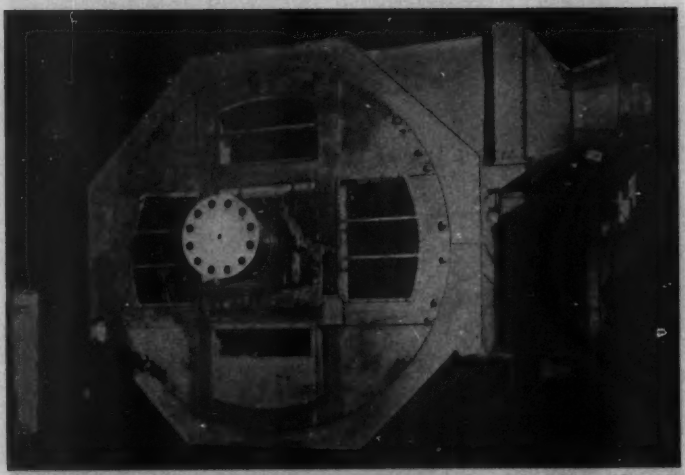


FIG. 7.—MOTOR OF 13,000 S.H.P. FOR THE BRITISH TANKER R.F.A. "OLNA".

FIG. 8.—PROPULSION MOTOR OF 13,000 S.H.P. IN THE BRITISH TANKER R.F.A. "OLNA".

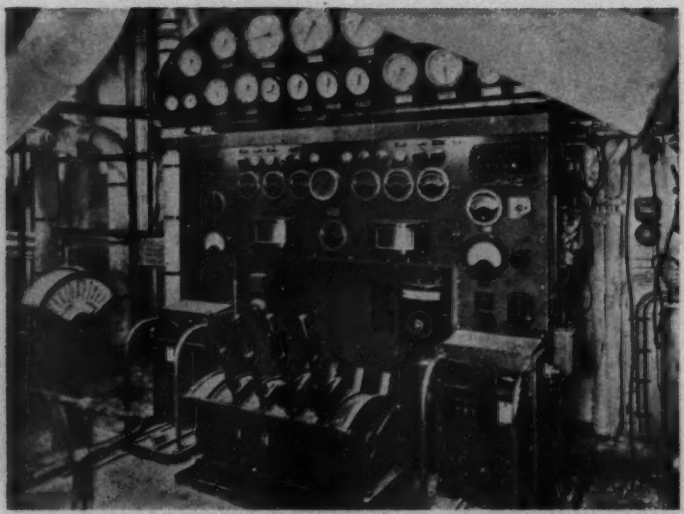


FIG. 10.—CONTROL PANEL FOR SINGLE-SCREW 13,000 S.H.P. DOUBLE-UNIT PROPULSION MOTOR IN BRITISH TANKER.

The main condenser is of the divided type so that, in case of a leaky tube, either side can be isolated and the tube plugged, with only slight loss of vacuum and without interruption of power.

Excessive weight has often been held as one of the disadvantages of electric drive, and, while this may have been true in the early days, the use of welding and fabrication generally has had a very marked effect. The following figures for this installation are therefore of special interest:—

Turbo generator	40.5 tons
Propeller motor	48 tons
Air coolers and fans	2.75 tons
Control gear	3.5 tons
	94.75 tons

The total weight (wet) of the complete power plant and auxiliaries is 405 tons, equal to 0.08 tons per shaft horsepower which compares favorably with either geared turbine or Diesel drive. Stopping tests in the fully loaded condition gave a time of 5.08 minutes from full speed to all way off the ship, and the distance travelled was 0.40 nautical mile.

It is of interest to turn now to the U. S. Maritime Commission single-screw tanker, type T2–SE–A1, of which 438 were built during the 6½ years ending in June last. Although modeled on similar lines to the *J. W. Van Dyke*, important changes, due to wartime restrictions, had to be introduced. For instance, the available supply of special steels for such a large program had to be taken into consideration and this influenced the choice of steam conditions, with a consequent reduction in efficiency as compared with the prototype. The turbo-generator rating is 5400 Kw., 3715 Rpm., 10 stages, 2370 volts, 62 cycles per sec., and steam conditions 435 lb. per sq. in. 720 deg. F. at throttle valve. The synchronous propulsion motor is of 6000 Shp. and 90 Rpm. normal rating but suitable for continuous duty at 6600 Shp. and 93 Rpm.

It speaks well for the liberality of the design of all the items comprising this equipment—namely, the boilers, steam turbine, alternator, motor, and control gear—to record that to meet wartime conditions many of these ships are running continuously at 8000 Shp. Operation is by the standard method already referred to and excitation is obtained from one of the 75 Kw. exciters coupled in tandem with each of the two 525 Kw. auxiliary D.C. turbo-generators, thus providing one standby exciter set.

Figs. 2 and 3, are views of a U. S. troopship equipment.

Opinions differ as to the necessity for arranging means for the automatic extinction of fire in the alternators and motors and British practice is not to make this provision: it is, however, the American rule and also that of some Continental firms. In these particular equipments carbon dioxide is released and the field switch opened in the event of a fire in the windings.

A method of arranging the cooling system in regard to the propulsion motor is shown in Fig. 4. The control-room panel of this installation is shown in Fig. 5. (See also Fig. 6, which shows a turbo-alternator.)

Developments in Germany. Pre-war Germany was leaning more and more towards Diesel-electric machinery with alternating-current transmission and was definitely electrically minded in her approach to propulsion matters. This trend was stated to be on the technical grounds of greater efficiency, but there were also other considerations. In the first place, orders were directed according to the district or factory it was desired to employ and, in the second place, edicts were issued that her merchant seamen were to have a regulated amount of leisure time while their ship was in port. The multiple-unit Diesel-electric plant, and to a lesser extent the turbo-electric plant, permitting as they did the shutting down of one unit for maintenance purposes without appreciable loss of speed, enabled this to be achieved. For instance, in a four-unit equipment, 91 per cent speed can be got from three units while the fourth receives attention.

It is interesting to recall that in 1935 a large-scale experiment was carried out by building three similar ships, the *Potsdam*, *Scharnhorst*, and *Gneisenau*. The last mentioned had geared turbines and the other two were turbo-electric.

The equipments have been fully described elsewhere and comparative operating results have also been published. Each main propeller motor is rated at 13,000 Shp., 130 Rpm. and turbo-generators and motors, contrary to normal practice, are constructed for an overload of 25 per cent. Steam conditions are 750 lb. per sq. in. at 842 deg. F., except that, on the *Potsdam*, which has Benson boilers, the pressure is 1200 lb. per sq. in., the temperature being the same. The *Gneisenau* and *Scharnhorst* both have Maierform hulls, but the *Potsdam* is of entirely different form, having a bulbous bow.

The mean operating results for the year 1937 were:—

	Gneisenau Voyages 5–8	*Scharnhorst* Voyages 6–9	*Potsdam* Voyages 6–9
Displacement, tons mean	22,600	22,900	20,440
Speed at sea (average), knots	19.82	20.0	19.7
Engine power, shaft horsepower mean	22,000	22,600	18,700
Days at sea	232	227	154
Boiler oil consumption (total), tons	38,439	36,619	37,882
Boiler oil at sea, tons	30,319	33,603	35,623
Boiler oil at sea, tons per day	156.7	147.9	154.05
Boiler oil lb. per Shp.-hr.	0.65	0.605	0.755
$\frac{\text{Boiler oil at sea, lb. per hr.}}{\text{Deadweight} \times \text{speed, tons} \times \text{knots}}$	0.069	0.065	0.06

French Ships. Although French installations are by no means numerous, they include the impressive, if ill-fated, *Normandie:* in her day the largest and most powerful liner afloat and, at that time, the holder of the Blue Riband for the fastest Trans-Atlantic crossing, with an average speed of 29.94 knots from east to west and 30.31 knots for the return journey. During this trip, condenser trouble necessitated one set having to be shut down, with a reduction in speed of only 2 knots; an operation which took three minutes to accomplish and which would have been quite impossible with geared turbine drive.

The plant consisted of four turbo-alternators each of 33,400 Kw. and 2430 Rpm., and if we add to these the auxiliary generating plant totalling 13,000 Kw., we have a floating power station with a total of nearly 150,000 Kw. The four propeller motors were each rated at 40,000 Shp. at 243 Rpm. and 6000 volts. At reduced speeds, such as would be required on winter schedules, one turbo-generator can drive two shafts, so that two turbo-generators can be shut down with a corresponding economy of the order of 8 to 10 per cent. Under these conditions the stationary sets can be overhauled, thus reducing lay-up periods to a minimum. Steam was supplied by water-tube boilers at 400 lb. per sq. in. pressure and a temperature of 680 deg. F.

The turbine had two cylinders, the high-pressure turbine having thirteen stages, and the low-pressure turbine was of double-flow design with three stages in each section. The rotor shafts were particularly stiff, their critical speeds being 30 per cent above the maximum running speed. Five tandem motor-driven exciter sets, with a separate exciter for each motor and alternator, left one set as a standby. During trials, a maximum speed of 32.2 knots with 165,000 Shp. at a displacement of 68,000 tons was recorded and fuel consumption on her maiden voyage was stated to be 0.71 lb. per shaft Hp.-hr. for all purposes. In succeeding voyages an average of 0.649 lb. per shaft Hp.hr. hour corrected to a calorific value of 19,000 Btu. per lb. of fuel oil was achieved.

Some Recent British Equipments. The present program of turbo-electric construction, in addition to a repeat of the S. S. *Rangitira* for cross-channel service between Christchurch and Wellington, falls into three groups as follows:—

Description	Number	Gross tons	Shaft horse-power	Propeller speed Rpm.
Bel ships	4	7,800	7,500	119
Tankers	3	13,000	13,000	122
Cargo liners	4	10,000	9,000	108

The steam conditions are as follows:—

Description	Steam pressure at stop valve, lb. per sq. in.	Steam temperature at stop valve, deg. F.
Bel ships	415	750
Tankers	425	740
Cargo liners	800	850

The equipment of the *Bel* ships follows orthodox lines, except in the control gear in which the reversing switches have been simplified in construction through the simple expedient of introducing a short time delay into the switching sequence.

The tankers are noteworthy in that they revive in a different form the system of transmission originally used in the M. V. *Tynemount* in 1912. The vessels are intended for high-speed operation at 16 knots under wartime conditions, but eventually the owners will be able to operate economically at a slower speed of approximately 12.5 knots. Consequently, they are provided with two turbo-generators, in conjunction with a double-unit motor on a single-screw or, in other words, two motors on one shaft. The two motors are identical in rating and at maximum power both are in operation, but at low powers only one need be used. Each alternator and motor forms an independent electrical circuit and the two turbo-alternators are in effect synchronized mechanically through the shaft of the double-unit motor (Fig. 7 and Fig. 8). A diagram of the appropriate wiring circuit is shown in Fig. 9, the control panel being illustrated in Fig. 10.

Maximum turbine rating is 5000 Kw., 4150 Rpm., 425 lb. per sq. in. gauge, 740 deg. F. total steam temperature and the voltage is 3000-3200 three phase. The motor is rated at 13,000 Shp. at 122 Rpm. Steam for feed heating is

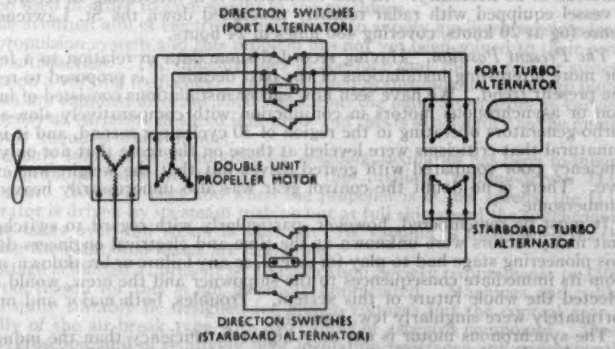


Fig. 9. Schematic Diagram of Main Circuits of Double Unit Motor and Two Turbo-Alternators of 13,000 s.h.p. Tanker

bled from the first and third stages. For maneuvering and speed control, five levers are provided as follows:—

(1) Direction lever.
(2) (*a*) Starting lever (port alternator).
    (*b*) Starting lever (starboard alternator).
(3) (*a*) Speed control (port turbine).
    (*b*) Speed control (starboard turbine).

When operating at reduced power, only the appropriate starting and speed levers, i.e. port or starboard, are used, but at full power two levers are moved simultaneously. If, when running with one alternator, it is desired to bring in the second set, it is only necessary to bring the speed lever momentarily to the slow position, go through the starting sequence with the incoming starting lever, and increase speed. This is done in a matter of seconds, and without elaborate synchronizing.

Full details of the cargo ships have not yet been released for publication and can be only briefly stated. They incorporate some advanced ideas and their performance will be watched with keen interest. Apart from the steam conditions, the chief items of note are that there is only one boiler and that the turbine has twin cylinders with reheat between cylinders. To provide a standby source of power in the extreme emergency of the boiler or the turbo-alternator being completely out of commission, a Diesel-driven auxiliary D.C. dynamo unit is coupled in tandem with an A.C. generator which will give sufficient power to the propeller motor to maintain steerage way on the ship. A very low fuel rate is anticipated and it will be interesting to compare the cost of fuel with that of oil engines of similar capacity, bearing in mind the relative prices of the two kinds of fuel.

Bridge Control. No installation of this type has yet been provided with bridge control, although one is now under construction in America. It is not a simple matter to arrange, and opinions differ as to the necessity. The time of greatest need would arise in an emergency calling for a "crash" stop in order to save precious seconds or even split seconds, but at all other times the normal telegraph system meets all requirements.

Gadgets have an awful habit of failing just when they are most needed. In fog or in dirty weather it is customary to reduce speed and if this practice, with the aid of radar, is maintained, emergency stops should be less frequent; but if, on the other hand, Radar leads to higher speeds, "crash" stops will continue to be necessary. In this connection it is interesting to record that a vessel equipped with radar recently proceeded down the St. Lawrence in dense fog at 20 knots, covering 480 miles in 24 hours.

The Present Position. Having recorded some data in relation to a few of the more interesting installations of the past decade, it is proposed to review the present trend. We have seen how earlier installations consisted of induction or asynchronous motors in conjunction with comparatively slow-speed turbo-generators operating in the region of 30 cycles per second, and it is not unnatural that criticisms were leveled at these on the score that not only was efficiency poor compared with geared drives but that the weight was excessive. There is no doubt the control gear was also unnecessarily heavy and cumbersome.

It must be remembered, however, particularly with regard to switchgear, that many factors were unknown at the time and electrical engineers during this pioneering stage had to play for safety, as any failure or breakdown, apart from its immediate consequences to the shipowner and the crew, would have affected the whole future of this system. Troubles, both major and minor, fortunately were singularly few, however.

The synchronous motor is not only higher in efficiency than the induction motor but, because it operates at unity power factor, a smaller alternator is possible. The advent of welding and fabrication results in a considerable saving in weight in every direction, including the motor, alternator, turbine, bedplates, and condensers. Auxiliaries such as exciters are also lighter and it

will be noted that the speed of the turbines in many of the examples quoted is of the order of 4000 Rpm.

It is probable that a further saving in weight is possible by a review of the permissible temperature rise. Motors and alternators have in the past been very conservatively rated for temperature rise, and sufficient advantage has not been taken of the fact that these sets almost invariably operate with water coolers. Recognition must also be given to the superior types of insulation and improved technique now available when reviewing this question.

Control gear similarly had to be liberal in design until experience had been gained in problems for which there is no parallel in land installations. Experience has shown that there is room for considerable reduction in the size and equipment and therefore in weight. The size of the control cubicle of the T2–SE–A1 tanker is an example of what can be achieved in this direction. For these 6600 Shp. equipments the whole of the control gear is mounted in a frame measuring 5 ft. by 5 ft. by 8 ft. and it is possible that this same control unit could handle considerably greater power. Reference has already been made to the fact that it is handling 8000 Shp. under wartime conditions.

There has been a tendency in the past to over-elaborate control gear and there is a strong case for simplification. Foolproof interlocks should only be employed where a false operation would cause damage to the plant. Where no damage but only temporary inconvenience would result, reliance must be placed on the intelligence of the engineer.

The employment of automatic regulators, unless they are of very simple construction, for reducing excitation at low speeds, is of doubtful value unless accompanied by a trimming of the alternator design. In practice, stability of the electrical system is maintained if the ratio of voltage to frequency remains constant and in some cases automatic regulators have been provided to maintain this ratio. It means that at reduced speeds less excitation is needed, but it is questionable whether this small gain in efficiency warrants the expense and complication.

On the other hand, whereas early designs without regulators gave a large degree of stability or torque margin to accommodate fluctuating loads in a seaway, the use of an automatic stability regulator does permit this margin to be reduced and smaller, lighter machines used. This practice has been followed in the United States and a simple type of regulator in conjunction with "amplidyne" control, which enables treble excitation to be applied, not only means a saving of weight and size but considerably reduces the time required for reversals. Dynamic braking to stop the propeller when reversing is also used in order to save weight in the machines.

The eventual aim of electrical engineers is to supply auxiliary services from the propulsion system and this problem has not yet been solved to their complete satisfaction. The difficulty lies in the fact that the propelling system works on a varying frequency. In the United States tankers, the cargo pump motors are supplied direct at 2300 volts from the main alternators at varying frequency as variation in speed is desired, and this is satisfactory as the pumps are only required when in port.

The only solution so far adopted for seagoing auxiliaries has been to mount a synchronous motor in tandem with the auxiliary turbine-driven D.C. generator. When in port and when the propeller is at low speeds, the D.C. generator is driven by its steam turbine but at full ship speed the synchronous motor takes over and steam is shut off. As vacuum is maintained on the turbine the rotational losses are negligible.

Voltage is limited at present by Classification Society requirements to 3300, and the reason for this is not generally appreciated. Machines and cables could quite suitably be designed for higher voltages but the control gear is usually of the air-break type with arc chutes of asbestos compound. These are hygroscopic and of relatively poor insulation resistance and consequently unsuitable for much higher voltages.

Oil-immersed switches were used in the early days but contacts breaking current in oil are only suitable for infrequent operation. When used for frequent switching rapid wear and pitting occurs and the result is cumulative.

The cable runs are comparatively very short and the saving in weight resulting from a higher voltage would be insignficant; nevertheless, Classification Societies would no doubt lend a sympathetic ear to any sound scheme if a higher voltage were preferred. In some equipments 5000 volts have already been used and the T. S. *Potsdam* employed 6000 volts.

It is not the purpose of this lecture to discuss the pros and cons of high-pressure high-temperature steam for marine plants as it is a subject entirely on its own and, furthermore, offers no advantages to turbo-electric plant which are not equally appropriate to geared drives. The object of such higher pressures and temperatures is reduction of the fuel bill, coupled with reduction in bunker space and weight. Generally speaking, the cost of fuel in a modern vessel is from 11 to 15 per cent of the total operating costs, from which it will be seen that in actual fuel costs a considerable reduction must be made to have any appreciable effect.

On the other hand, for tankers and dead-weight cargo vessels, every ton saved on the machinery represents a corresponding increase in cargo capacity: for bulky cargoes space is the main consideration and—since volume usually follows weight—the importance of these is apparent. The electric drive enables the motor to be placed abaft the cargo space, and, besides dispensing with intermediate shafting and eliminating its weight and that of its accoutrements, this arrangement also does away with the tunnel or tunnels which occupy valuable cargo space. Frictional losses in the intermediate shafting are of the order of 1 to 7 per cent of the power transmitted.

Savings in cost of fuel and gains in other directions already mentioned which will result from higher steam pressures and temperatures, must be offset by increased capital and maintenance costs, not forgetting the availability factor, moisture content of the steam, and increase in scantlings, and therefcre weight, of plant, steam pipes, etc. Taking all factors into account, it will probably be found that the most economic figure lies between 600 and 900 lb. per sq. in.

The combustion or gas turbine is now a practical means of propulsion and a Continental firm has a 10,000 Kw. set on order for a power station with a guaranteed thermal efficiency of 28 per cent. Such a high efficiency requires a large heat exchanger and it is claimed that, up to an efficiency of 26 per cent, the weight with a single-stage turbine is less than the corresponding weight of a steam plant. A 4000 Kw. set has been supplying the City of Neuchatel, Switzerland, since early in 1940.

A disadvantage for marine work would be the poor performance at low loads, but nevertheless investigations are proceeding and in the United States an experimental combustion turbine for marine service has undergone a series of tests. The development of a reversing combustion turbine will involve complications which may well be avoided very simply by adopting electric drive which operates with a unidirectional prime mover.

Experience has shown that marine engineers with no special electrical qualifications have no difficulty in operating electrically propelled ships and, in fact, one operating company with a fleet of these ships prefers not to carry any electricians. Nevertheless it is preferable that the engineers should have sufficient electrical knowledge at least to carry out emergency repairs, however, and it is the practice of most owners to allow their engineers to stand by at the electrical works during the construction of new machinery. Consideration may have to be given to an Electrical Endorsement of Engineer's Certificates of Competency in the same way that Diesel Endorsements are now granted.

Experience has shown that maintenance charges on the electrical side are practically non-existent. For example, in the case of two sister ships each with four screws, 19,000 Shp., published records show that in one of them, in eleven years of service of a very arduous character, the total cost of upkeep was £109, and for the other ship, over a period of approximately nine years, it was £576.

Changes in the national finances as an outcome of the war necessitate review of our pre-war prejudices. Unfortunately, the problem is complicated by the soaring cost of home-produced fuel but if we must use oil because it is

cheaper and more convenient than coal, we must at any rate see to it that we use as little as possible. Furthermore, if we are to regain our pre-war supremacy and national earnings from shipping and freight-carrying it is imperative that our ships attain the optimum of efficiency and minimum running costs. Turbo-electric drive will certainly not be the solution to every requirement, but will merit very serious consideration for ships of above, say, 3000 Shp. but more seriously from about 5000 Shp. upwards.

LUBRICATION CONSIDERATIONS IN GEAR DESIGN.

This article was prepared by the Technical Division, Industrial, Socony Vacuum Oil Co., and is reprinted from "Machinery" for May, 1947. It discusses the factors involved in selecting proper lubrication for various gear designs.

Gears will not operate satisfactorily unless a sufficient amount of lubrication is maintained between the surfaces of the meshing teeth. The very nature of gearing requires that provision for lubrication be considered early in the design stage. Power cannot be transmitted efficiently, nor the life of the gears extended to the economic maximum, if direct metallic contact of meshing gear teeth is permitted to absorb useful power and cause rapid tooth wear or early surface fatigue failure. On the other hand, even a combination of the correct gear design and a lubricant of the correct type does not guarantee that operation will be trouble-free, because some gears are inherently difficult to lubricate—the teeth may vary in hardness, degree of finish, and alignment—and the actual service conditions may be different from the assumed design service conditions.

Film-Forming Ability.

Like bearings, gears may be lubricated by a full film of oil that is drawn into the space between contacting surfaces, or by an extremely thin film of oil under so-called boundary lubrication conditions. The type of contact and relative motion between mating teeth decide which of these conditions exists.

For example, consider the nature of contact between two spur gear teeth of customary form, as shown in Fig. 1. At top, the driving teeth are coming into mesh, and make their first contact with the trailing tips of the driven teeth. As the mating teeth go further into mesh, both a rolling and sliding action take place. The length of the distance traveled is relatively much shorter for the driving teeth, representing roll, than for the driven teeth, representing slide.

The lines of contact become longer as the action proceeds. Finally, the teeth reach the center of mesh, and contact between them occurs only at the pitch line. The action is momentarily one of rolling. As the teeth begin the period of disengagement, rolling gives place to a combined sliding and rolling action but, in this instance, the longer distance of travel occurs on the driving teeth rather than on the driven teeth.

The combined sliding and rolling action promotes the formation of separating oil films, and hence of itself is a favorable factor in full fluid-film lubrication. When this fundamental action is applied to gear teeth of finite width, so that tooth surfaces rather than theoretical tooth outlines are involved, conditions may be different from the ideal just described. It will be found, as a result, that the several types of gears vary in their inherent ability to establish and maintain the substantial film of lubricant that is desirable for the most effective lubrication; this is shown by Tables 1 and 2.

The action of the mating gears shown in Table 1 produces a situation that is illustrated by Fig. 2. The line of tooth contact is perpendicular, or almost so, to the direction of the film-forming action, so that oil introduced into the teeth as they start engagement is drawn positively between them, tending to form a separating film of substantial thickness and to prevent direct metallic contact. There is then no reason for film rupture or failure as far as the type of gear is concerned. However, boundary lubrication may be encountered

Fig. 1. Contact between Meshing Gear Teeth is a Combination of Rolling and Sliding. During the Period of Engagement, the Length of the Distance Traveled (Shown by Heavy Line) is Greater for the Driven than for the Driving Gear; the Reverse is True as the Teeth Leave the Center of Mesh to Start the Period of Disengagement. The Rolling and Sliding Action Promotes the Formation and Maintenance of Separating Oil Films

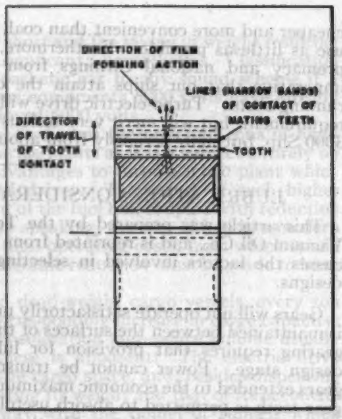

DIRECTION OF FILM FORMING ACTION
LINES (NARROW BANDS) OF CONTACT OF MATING TEETH
DIRECTION OF TRAVEL OF TOOTH CONTACT
TOOTH

Fig. 2. The Gears Shown in Table 1 are Favorable to the Formation and Maintenance of Full Fluid-lubricant Films between Meshing Teeth because They Satisfy Conditions Shown Here. The Direction of the Film-forming Action is Perpendicular, or Almost so, to the Line of Contact between Meshing Teeth

when unit loadings increase sufficiently to rupture the oil film, or when the speed is so low as to hinder or prevent an ample flow of oil to the active tooth areas.

In the gears shown in Table 2, point contact or marked side-slide is present, and these conditions are not favorable to full fluid-film lubrication. Boundary lubrication is usually the best that can be attained.

Because of the difficult lubrication conditions inherent in helical gears operating on crossed shafts, including non-throated worm-gears, such gears are customarily designed to handle low tooth loadings. Even then, lubrication will be of the boundary type, and unless the correct oils are used, it is doubtful if even boundary conditions can be maintained sufficiently to avoid excessive tooth wear.

The difficult conditions present in single-throated worm-gears are partially overcome by making the gear itself of friction-reducing bronze. This, together with the fact that several teeth are in contact simultaneously, helps to overcome the difficult conditions. Elastic deformation of bronze tooth surfaces increases the contact area and reduces unit pressures.

The use of hypoid gears for industrial purposes need not be accompanied by lubrication difficulties. In contrast with automotive practice, space for accommodation of the hypoid gear set is ordinarily not unduly restricted. Consequently the gears can be made relatively large, with reasonable unit loadings, in order to eliminate the extremely high pressures that tend to force even the strongest, most adhesive oils out of the spaces between mating teeth. A correctly designed hypoid gear for industrial use should be considerably less difficult to lubricate than a single-throated worm-gear, but somewhat more difficult than a spiral bevel gear.

Summing up then, it is found that of the gear types in widest use, spur gears, helical gears on parallel shafts, herringbone gears, bevel gears, and spiral bevel gears are inherently favorable to the establishment and maintenance of full fluid films. Double-throated worm-gears, in light of all present knowledge concerning them, are also apparently favorable to lubrication. Helical gears on crossed shafts and non-throated worm-gears are difficult to lubricate, especially if they are heavily loaded. Single-throated worm-gears and hypoid gears with considerable shaft offset are not favorable to full fluid-film lubrication, and boundary lubrication conditions will ordinarily prevail.

Obviously, inherent capacity for easy or sufficient lubrication is not the sole criterion in the selection of a gear type. The designer has to consider other factors as well, such as available space, quiet operation, permissible tooth loading, load distribution over more than one tooth, angularity between driving and driven shafts, and speed ratio—all of which affect the choice of gear.

In addition, the designer has a direct, although perhaps limited interest in manufacturing processes and control. Gear steels should be specified that are of known quality and are produced with close control over composition and uniformity. His interest should extend as well to consideration of the uniform reaction of the material to forging, machining, and heat-treating operations. Uniformity of raw materials and manufacture contributes to uniformity of product, a prime requisite in successful gear operation.

Designed tooth loadings may be exceeded in operation because of variations in surface smoothness and hardness or because of tooth misalignment. Unit pressures may become so high that oil films are punctured or entirely squeezed away, and harmful metallic contact results. Also, when surface smoothness and hardness are not uniform, pitting is likely to occur; such pitting may grow progressively worse when conditions are severe and cannot be prevented even by the use of a correct, high-quality lubricant.

A higher degree of surface finish on the gear teeth may be necessary. Pronounced surface irregularities cut through oil films and cause galling of the surface metal.

Lubricating the Gear Set.

To be of practical use, a gear must be supported on a shaft, which in turn is carried in some form of bearings. This suggests also the use of a housing or support for the bearings. The latter may be very elementary or may be

Table 1. Gears Favorable to Formation of Full Fluid Film

SPUR GEARS

Spur gear teeth are cut straight across the face of the gear blanks, and the contact with respect to the mating surfaces is a line which may be widened, under load, to a narrow band that extends across the face of the teeth. This line of contact, being at right angles to the film-forming action, sweeps oil into the pressure area and is responsible for the ability of spur gears to maintain fluid films of oil between contacting surfaces.

HELICAL GEARS ON PARALLEL SHAFTS

If a helical gear is considered as an infinitely large number of laminations, each offset a small amount from its neighbor, the theoretical contact is a curved, slanted line composed of the contact points of adjoining laminations. Under load, this slanted line becomes a band extending across the face of the gear teeth, which is so nearly perpendicular to the line of film formation that helical gears on parallel shafts are favorable to full fluid-film lubrication. This also applies to herringbone gears, as the individual gear is basically two helical gears on the same gear blank.

BEVEL AND SPIRAL BEVEL GEARS

Since the action of the teeth of bevel gears is like that of spur gears, they also are favorable to full fluid-film lubrication. A spiral bevel gear can be regarded as a bevel gear whose teeth have been twisted to form a spiral. The line of tooth contact is nearly perpendicular to the line of oil-film formation and this creates a condition favorable to fluid-film formation.

DOUBLE-THROATED WORM-GEARS

In this relatively new development in gearing there is no rolling action between mating gear teeth; the action is one of sliding only. Moreover, the teeth appear to make band contact radially. This, together with the fact that the construction causes several teeth of the worm and gear to be in mesh at any instant, suggests that this type of gear should be favorable to good film formation. While the entire subject of double-throated worm-gears is still under considerable investigation, experience to date indicates that effective lubrication is possible.

Table 2. Gears Less Favorable to Full Fluid-Film Formation

HELICAL GEARS ON CROSSED SHAFTS

When mating helical gears operate on crossed instead of parallel shafts, the theoretical contact is always a point. While the point may be enlarged to a small circular area by deformation of the metal under load, the pressure is nevertheless high during contact. Furthermore, as with all gears mounted on crossed shafts, side-slide is present. Even though basic sliding and rolling take place during tooth contact, the resulting film-forming tendency is hindered by both the point contact and the side-slide.

SINGLE-THROATED WORM-GEARS

In a single-throated worm-gear, tooth contact is on a curved line. The sliding and rolling action of the gear teeth is radial, as in spur gears, but the rotation of the worm imposes side-sliding, and because the speed of the worm is high compared to that of the gear, side-slide predominates. The resulting line of action of the net side-slide almost coincides with the line of tooth contact. Full fluid films of lubricant cannot be maintained easily under these circumstances, and conditions are therefore not entirely favorable to the formation of full fluid films.

NON-THROATED WORM-GEARS

The teeth of a non-throated worm-gear are cut angularly across the face of a blank. They are not curved and do not envelop the teeth of the worm. Also, since they are a form of helical gears mounted on crossed shafts, point contact prevails, the contact pressure is relatively high, and side-slide is present; hence these gears are not favorable to lubrication.

HYPOID GEARS

The line of contact between mating teeth of hypoid gears is a curved, slanted line, somewhat like that found with spiral bevel gears. When the shafts of pinion and gear are only slightly offset, conditions are similar to those prevailing with spiral bevel gears, and hence are favorable to lubrication. As the amount of offset increases, side-slide progressively assumes greater importance and the lubrication problem becomes more difficult.

Fig. 3. With This Method of Lubrication, the Main Supply of Oil is Kept Relatively Free of Agitation and Churning. The Lower Half of the Gear is Encased by a Metal Cover, into which Oil from the Reservoir Enters through Suitable Holes. The Rotation of the Gear Carries the Oil Upward and Throws it against the Upper Half-casing, where it is Cooled by Radiation. The Oil Thrown by the Gear is Caught in Troughs on the Inside of the Upper Casing, and is Fed by Gravity to the Bearings. Drain Holes in the Bearing Seats Return the Oil to the Reservoir

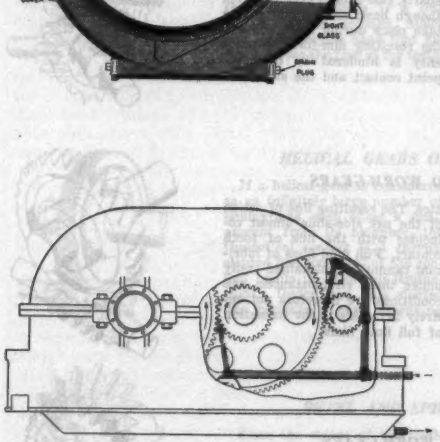
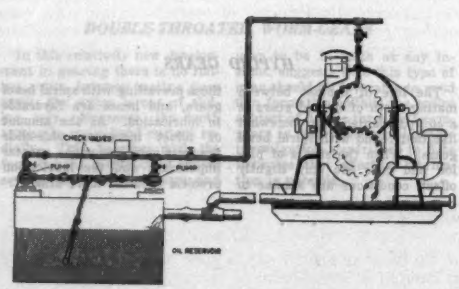


Fig. 4. In Forced-circulation Oiling Systems Such as This, it is Important that the Spray of Oil be Directed at the Teeth Just as They Start Engagement

Fig. 5. Because the Gears of This Pinion-stand Operate in Either Direction, Provision is Made to Feed Oil to Both Sides of the Meshing Teeth. Oil, Pumped from a Reservoir, is Flooded over the Upper Gear. Due to the Rotation of the Gear, the Oil is Diverted to a Baffle Plate, which in Turn Guides it to the Incoming Teeth of the Lower Gear. Thus, Irrespective of Direction of Rotation, Oil is Fed to the Teeth of Both Gears on the Approaching Side of Mesh



developed into a more or less elaborate structure, entirely enclosing the gears and acting at the same time as a sump or reservoir for holding the gear lubricant.

The design of shafts, bearings, and bearing supports or housings is a matter of the utmost importance. They must be developed along with the gear design itself, or else the gear set is likely to be improperly lubricated.

With enclosed gearing, it is incumbent on the designer to carefully assess the heat generated during operation and provide a sufficient heat transfer surface in the gear-case or oil-cooler to maintain moderate operating temperatures. Lubricating oil that becomes excessively thinned due to high temperatures is more easily squeezed from between the teeth, and metal-to-metal contact results. Ignoring any heat rise due to external causes, the total amount of heat generated is in practically direct proportion to the horsepower transmitted, whereas the physical size of the gear set is increased at a lesser ratio. Hence, the total natural radiation from a large case is relatively of less magnitude than from a small case, and means of forced cooling of the lubricating oil may have to be provided for the larger unit.

The exact nature of the proposed lubricating system also requires consideration. When splash or bath lubrication is used, the normal oil level must be carefully determined. If the oil level is too high, unnecessary churning of the lubricant brings about high temperatures, which thin the oil excessively and accelerate oxidation. This condition is encountered frequently in small, high-speed sets, and indicates that the design should provide for the correct oil level and for the optimum radiating capacity of the gear housing or casing. On the other hand, large splash-oiled gear sets may also subject the lubricant to excessive churning. This condition can be avoided by encasing the lower half of the gear in a metal cover having holes which admit oil in limited quantity from the main supply. The gear dips into a relatively small quantity of oil, and the main supply is kept free of agitation and churning. This is illustrated in Fig. 3.

With splash lubrication, it is also wise to provide a settling or sediment chamber by damming or baffling off that part of the oil reservoir where the bulk of the oil drops back to the main supply. The oil in such a reservoir lies comparatively still and solid impurities settle to the bottom; the relatively clean oil then flows slowly over the dam to the main body of lubricant. Magnetic drain plugs at strategic points in the gear-case assist in removing iron and steel particles whose continued presence would cause wear by abrasion, or perhaps would accelerate oxidation of the lubricant by catalytic action.

Circulation oiling, with the lubricant sprayed on the gears (as illustrated in Figs. 4 and 5), encourages radiation of heat from the oil to the air, and normally eliminates the heat developed due to churning of the oil in a bath or splash system. Part of the spray strikes the sides of the case and is cooled as a result. In a circulation system, cooling by a separate oil cooler is more easily arranged. Such a system also permits the use of oil filters, a practice that assists lubrication by removing harmful abrasives and sludge from the lubricating oil.

The point of application of oil to the meshing teeth is very important in forced-circulation systems. In general the spray of oil should be directed at the teeth just as they start engagement, as shown in Fig. 4. Likewise, if the gears are required to operate in either direction, spray provisions must be made for both conditions. Frequently, deflecting troughs or baffles will accomplish the same purpose, as in the pinion-stand shown in Fig. 5.

Breather openings or vents should be provided for enclosed gear sets. Any build-up in pressure that tends to cause leakage outward will thus be avoided. Venting also helps cool the gear-case and assists in the removal of potentially harmful water vapor. A means of venting is almost imperative when a gear set is expected to operate on an intermittent basis, as the alternate heating and cooling are favorable to the accumulation of moisture by condensation of airborne water vapor. Any water that accumulates in the gear-case will tend to cause rusting of the gears and interior of the housing and the formation of

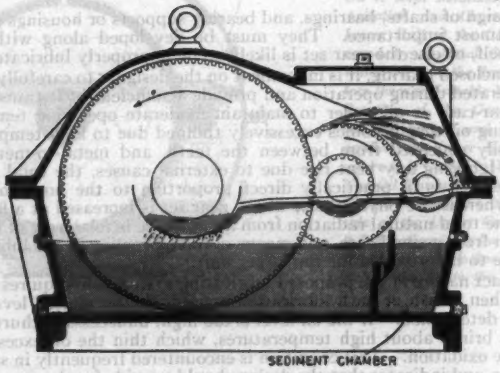


Fig. 6. In This Double-reduction Splash-lubricated Gear Set, Oil Splashed from the Gears Drains into a Trough in the Housing and Thence to the Bearings

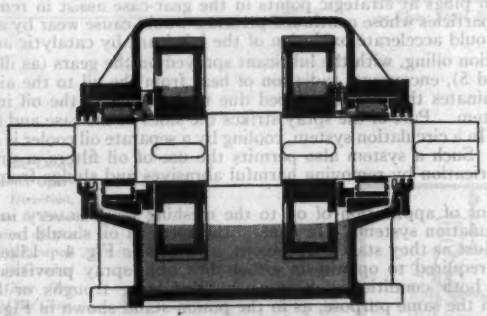


Fig. 7. Splash-lubricated Gear Set in which Pockets are Cast in the Gears on the Low-speed Shaft for the Purpose of Carrying Oil to the Bearings

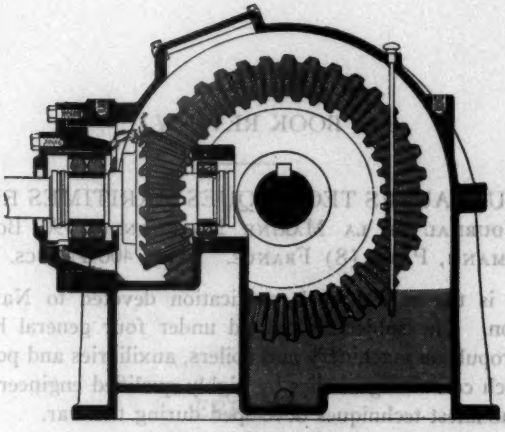


Fig. 8. Splash-lubricated Bevel-gear Set in which the Bearings are Lubricated by Direct Splash from the Gears

emulsions and sludges—conditions that must be avoided as much as possible in the interests of gear and lubricant life.

Bearings usually are lubricated by the same oil that is used for the gears themselves. This is a relatively simple procedure when circulation by pump or gravity means is utilized. In splash-lubricated sets, part of the oil thrown from the gear is led to the bearings. Troughs, pockets in the gears, and direct splash from the gears are common methods by which bearings are lubricated, as shown in Figs. 6, 7, and 8.

There is little the designer can do for the lubrication of open gearing other than to design the gears for the load, speed ratio, and any other requirements involved, and supply adequate supports in the form of shafting and bearings. He can, however, specify the installation of a mechanical force-feed lubricator or a simple drop-feed oil-cup that will permit the application of small but regular feeds of a lubricant having characteristics valuable in boundary film lubrication. The force-feed lubricator, because of its positive action, is to be preferred if a suitable mounting and drive can be arranged.

The problem of gear lubrication resolves into one of lubricant selection and intelligent maintenance of gears and lubricating oil. Fortunately, oil refiners are keenly aware of the lubrication needs of modern gearing of both the enclosed and open types. They have available a variety of proved lubricants and lubrication engineers who are qualified by training and experience to consult with the gear user and advise him on lubricant selection and maintenance.

BOOK REVIEW.

"NOUVEAUTES TECHNIQUES MARITIMES EN 1946".

THE JOURNAL DE LA MARINE MARCHANDE, 190 BOULEVARD HAUSSMANN, PARIS (8) FRANCE. PRICE 400 FRANCS.

This is the only French publication devoted to Naval Construction. The subject is treated under four general headings; hull, propulsion machinery and boilers, auxiliaries and port activities, each containing studies by highly qualified engineers dealing with the latest techniques developed during the war.

Carefully prepared and abundantly illustrated, this publication contains 216 pages and is the result of efforts of engineers and members of industry interested in Naval Construction.

ERRATA.

On page 140, second paragraph, of paper entitled "Influence of Arctic Operations on Future Ship Design", by Captain R. B. Lank, Jr., U.S.C.G., in the May, 1947, issue of the JOURNAL, the name *Edisto* should be changed to *Eastwind*.

ASSOCIATION NOTES.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a

very fine, dignified insignia. It is one-half inch in diameter. The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

Changes in Council.

The following appointments to membership on the Council have been made, all due to detachment from duty in Washington of those replaced:

Captain D. L. Hull, U. S. N., in place of R. Adm. N. L. Rawlings, U. S. N.

Captain Robert L. Swart, U. S. N., in place of R. Adm. S. S. Kennedy, U. S. N.

Commodore H. C. Shepheard, U. S. C. G. R., in place of Captain R. E. Coombs, U. S. C. G. R.

Scholarship.

The Society has awarded the scholarship recently established by it at Webb Institute of Technology to John R. Alexander, Jr., son of Captain John Richmond Alexander, U. S. N. R.

Membership.

The following have joined the Society since the publication of the May, 1947, Journal:

Naval.

Anderson, Walter H., Ensign, U. S. N. R.,
1021 Elm Ave., Sanford, Fla.

Boehner, Herbert C., Lieut. Commander, U. S. N.

Brand, Carl R., Commander, U. S. N. R.,
Staff Headquarters, 17th Naval District, Kodiak, Alaska.

Brooks, R. L., Lieut. Commander, U. S. N.,
707 Greythorne Road, Wynnewood, Pa.

Bryan, Roland Towler, Lieut. j.g., U. S. N.,
Research & Development Department, The Babcock & Wilcox Company, Box 835, Alliance, Ohio.

Carmody, Maurice F., Ensign, U. S. N. R.,

Carr, James E., Lieut., U. S. N. R., Ret.,
259 South 44th St., Philadelphia, Pa.

Chambellan, R. E., Lieut., U. S. N. R.,
537 St. Pauls Ave., Cliffside Par, 10, N. J.

Crecca, John D., Captain, U. S. N.,
Board of Inspection & Survey, Navy Department, Washington, D. C.

Dunne, Roye, Lieut. j.g., U. S. N. R., Assistant to General Manager, Bull SS Line, Pier 9, Corpus Christi, Texas
Mail 1001, Ralston Ave., Corpus Christi, Texas.

Gerech, Walter T., Lieut., j.g., U. S. N.,
% D. S. Jenkins, Annapolis Boulevard, Annapolis, Md.

Gorman, Donald V., Ensign, U. S. N.

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